# THE GAS ENGINE

BY

CECIL P. POOLE

EDITOR OF POWER AND THE ENGINEER

AUTHOR OF

"THE WIRING HANDBOOK," "DIAGRAMS OF ELECTRICAL CONNECTIONS,"
"DESIGNS FOR SMALL DYNAMOS AND MOTORS," ETC.



1909

HILL PUBLISHING COMPANY

505 PEARL STREET, NEW YORK

6 BOUVERIE STREET, LONDON, E. C.

*Power and The Engineer — American Machinist — The Engineering and Mining Journal*

Pr





*Hill Publishing Company, New York, U. S. A.*

# PREFACE

THIS book is not intended as a complete treatise on the subject. The object of the author is to present the principles governing the salient features of gas-engine construction and operation in as simple a manner as possible, and to that end academic discussions of the characteristics of gases and of hypothetical heat-energy cycles, of the character commonly found in text-books, have been avoided. Since the pressures, temperatures, and energy transformations which occur in a gas-engine cylinder cannot be adequately explained without the use of algebraic equations which appear complex to a beginner, such equations have been employed in that connection, but their use has been restricted to a single chapter. This chapter may be omitted without sacrifice by readers who wish merely general, rudimentary information, but not by real students of the subject.

C. P. P.

NEW YORK, January, 1909

# CONTENTS

## CHAPTER IX



## CHAPTER X

## ERRATA

On page 73, in the second line of the first paragraph, the words "and pressure" should be omitted.

On page 74, in the third line of the last paragraph, the word "pressure" and the comma after the word "temperature" should be omitted.

# I

# ELEMENTARY PRINCIPLES

## The Working Medium

Gas and oil engines differ from other forms of heat engine chiefly in that the pressure which gives the engine its power is produced within the cylinder by the combustion of the gas or oil. For the operation of all other heat engines the working medium (steam, hot air, etc.) is raised to a pressure much higher than that of the atmosphere before it is delivered into the cylinder; after entering the cylinder, the working medium is expanded to a low pressure, the expansion driving the piston forward. The working medium of the gas or oil engine is delivered to the cylinder at about atmospheric pressure and there compressed and ignited; the rise of temperature produced by the combustion causes a corresponding rise of pressure and the high-pressure gases are then expanded behind the piston in the same manner that steam expands in a steam-engine cylinder, and with similar results.

In order to burn anything, no matter how inflammable it may be, it is necessary for oxygen to be brought into contact with the substance to be burned, because combustion is nothing more than the union of oxygen particles with the combustible particles of the substance "burned," under the influence of heat. Air, which consists of oxygen and nitrogen, is the only free source of oxygen and is therefore universally used to supply the oxygen required for combustion of any kind. Hence, the gas or oil used as fuel in an engine is always mixed with air either immediately before it enters the engine cylinder or immediately afterwards. The proportion of air to gas or to oil is of great practical importance; if too little air is supplied, combustion is slow and incomplete, and if the proportion of air is too large, the inflammability of the mixture is reduced and combustion is retarded. Again, it is important that the mixing of the air and fuel should be thorough; otherwise some of the gas or oil either will not be burned at all or will burn too late to do much good in the way of producing pressure behind the piston.

Gas and oil burn quietly in the open air chiefly because the gases produced by the combustion can expand as rapidly as they are formed, having

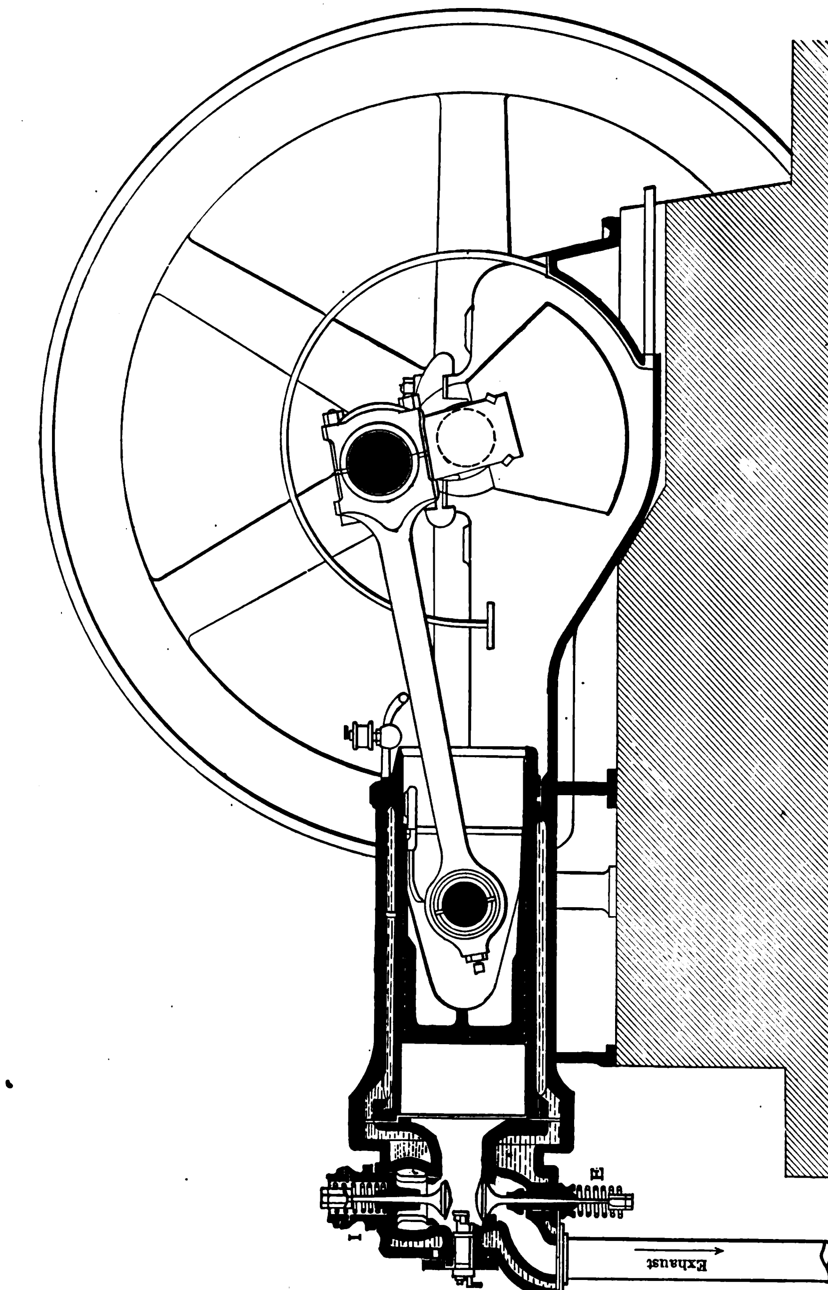



FIG. 1.—SINGLE-ACTING GAS ENGINE WORKING ON THE FOUR-STROKE CYCLE.

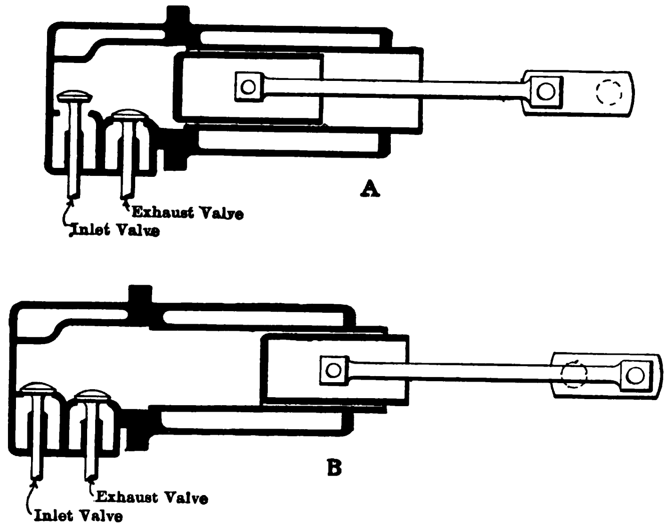
the whole universe into which to expand and being restrained only by the moderate pressure of the atmosphere. Moreover, the atoms of gas or oil are not mixed intimately with the atoms of oxygen in the air before burning begins, so that the combustion is very gradual. When gas and air or oil vapor and air are thoroughly mixed and highly compressed in a closed vessel, however, igniting the mixture will produce almost instantaneous combustion of the whole mass, resulting in a sudden rise of temperature and pressure amounting practically to an explosion. This is what happens in the cylinder of a gas or oil engine when conditions are right. The explosion of the mixture occurs when the piston is at one end of its travel, the mixture being compressed in the clearance space between the piston and the near cylinder head; and as soon as the crank passes the dead center, the burned and burning gases expand, forcing the piston away from the end of the cylinder. At the end of that stroke, the spent gases are exhausted into the atmosphere, just like the expanded steam in a non-condensing steam-engine cylinder.

## The Four-stroke Cycle

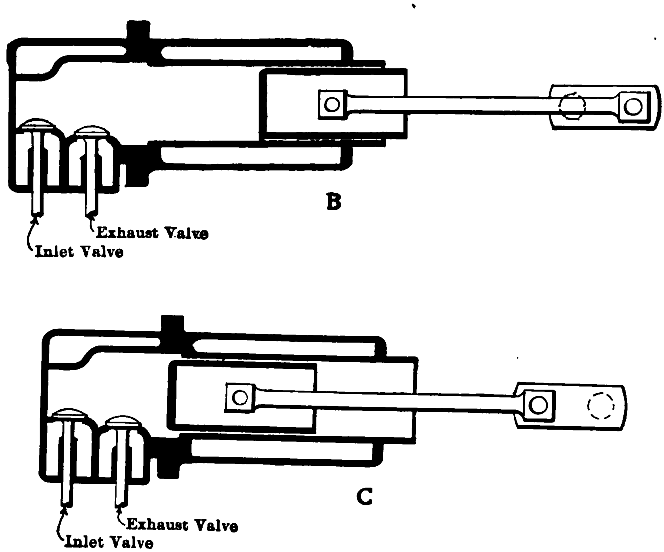
There are two general classes of gas and oil engines; one works on the four-stroke cycle and the other on the two-stroke cycle. All small engines of both classes are single-acting: the piston is commonly of the trunk type, as illustrated in Fig. 1, and the combustion of fuel occurs in one end only of the cylinder.

The four-stroke cycle is more widely used than the two-stroke, for reasons which will be explained later on. The cycle comprises five events, namely: Admission, compression, combustion or explosion, expansion and exhaust. In this type of engine the charge is taken into the cylinder at atmospheric pressure, and is therefore necessarily drawn or "sucked" in by the piston of the engine, acting for the time as a pump piston. This occurs during one out-stroke of the piston (from position *A* to position *B*, Fig. 2), during which the inlet valve is held open either by the valve gear or by the atmospheric pressure. At the end of this stroke, commonly called the "suction" stroke, the inlet valve is closed, and when the piston comes back on the return stroke (*B* to *C*, Fig. 2) it compresses the cylinderful of mixed air and gas (or air and oil vapor) into the clearance space, which is relatively much larger than in a steam-engine cylinder.

When the piston has reached the end of the compression stroke, and while the particles of mixed air and fuel are compressed into intimate contact, the mixture is ignited and burns explosively, as already explained, producing a sudden rise of pressure behind the piston. This enhanced pressure drives the piston forward on its power stroke (*C* to *D*, Fig. 2*a*), during which the pressure is gradually reduced by the expansion of the

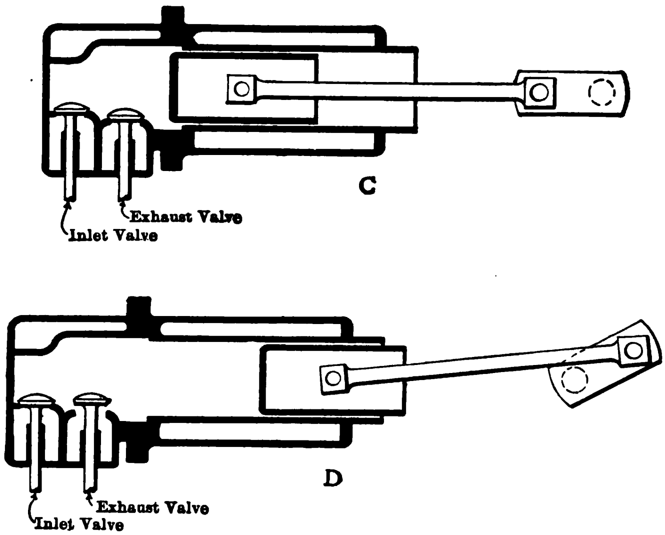



THE SUCTION STROKE.

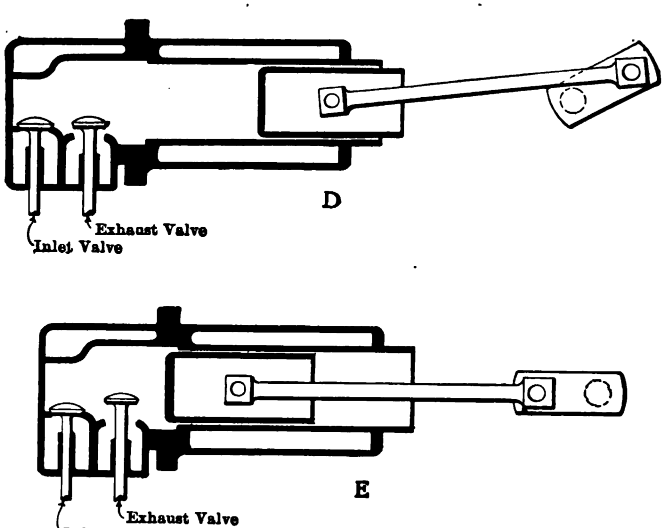



THE COMPRESSION STROKE.

FIG. 2. — ILLUSTRATING THE FOUR-STROKE CYCLE.




THE EXPANSION STROKE.




THE EXHAUST STROKE.

FIG. 2*a*.—ILLUSTRATING THE FOUR-STROKE CYCLE.

hot gases, the valves remaining closed, of course, until the stroke is almost completed; then the exhaust valve is opened by the valve gear and the burned gases allowed to expand through the exhaust port down to atmospheric pressure. On the return stroke (*D* to *E*, Fig. 2*a*), the piston drives the remaining hot gases out of the cylinder, except what remain in the clearance space at the completion of the stroke (position *E*).

It is evident from the foregoing that the five events in the engine cylinder occur during four strokes of the piston: the suction stroke, compression stroke, expansion or power stroke, and the exhaust or expulsion stroke; hence the term "four-stroke cycle." Fig. 3 is an indicator-diagram made

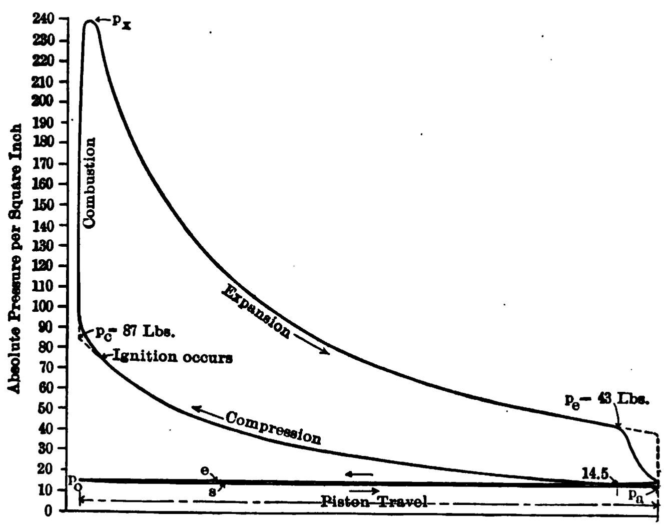



FIG. 3. — INDICATOR DIAGRAM FROM A GAS ENGINE WORKING ON THE FOUR-STROKE CYCLE.

by a gas engine working on the four-stroke cycle. The line *s* from $p_o$ to $p_a$ was traced by the indicator during the admission or suction stroke of the piston, and lies a trifle below the line *e*, which is at practically atmospheric pressure. The reason for this is that the piston, moving away from the cylinder head, forms a slight vacuum behind it before the fresh mixture begins to enter the cylinder, and this partial vacuum is maintained throughout the remainder of the suction stroke. Consequently the entering charge is at a slightly lower pressure than that of the outside atmosphere, as shown by the admission line *s* on the diagram. This is necessary, of course, because if the pressure within the cylinder were not lower than that of the atmosphere the mixture of air and gas would not enter, unless previously compressed to a pressure higher than the atmospheric pressure.

The degree of vacuum required to draw in the charge depends on the resistance offered to the charge by the passages and inlet-valve port through which it reaches the cylinder, and, in order to keep down the work done in "sucking" the charge through these passages and port, their area is made as large as practicable and all bends are of as large radius as constructional considerations will permit. On the diagram here reproduced the admission pressure was 13½ lbs. (inaccuracy in redrawing it has made it appear higher); the degree of vacuum, therefore, was 1.2 lbs. per square inch or 2.43 ins. of mercury.

The curve marked "Compression" shows the rise of pressure produced by compressing the mixture in the cylinder during the return piston stroke. If the mixture had not been ignited until the compression stroke was completed, the curve would have been continued to the point $p_c$, as indicated by the dotted extension of the curve, and if ignition had then occurred, the combustion line would have started abruptly upward, as indicated by the vertical dotted line. But it has been found advisable in practice to ignite the mixture just before the end of the compression stroke, and that is what was done in this case, producing an upward change in the compression curve at the point marked "Ignition occurs." The reasons for igniting the mixture before the piston completes the compression stroke are fully explained in the chapter on Ignition.

It will be noticed that the lower part of the combustion line is strictly vertical and that the line leans over slightly toward its upper end. That is due to the fact that combustion was not instantaneous, but continued after the piston had started forward on its power stroke. Absolutely instantaneous combustion is not obtained in an actual engine because of the impossibility of getting a perfect mixture and inflaming the whole of it at the same instant.

The expansion curve of the diagram will be recognized as very similar to the expansion curve of a steam-engine indicator diagram. It drops more rapidly, however, than the curve of steam expansion, and at the point $p_e$ its direction changes rather abruptly; this is due to the fact that the exhaust valve opens before the piston completes the expansion stroke, which is necessary in order to give the hot gases time to expand to atmospheric pressure before the piston starts back on the return (expulsion) stroke, and thereby avoid excessive back pressure during the first part of that stroke. The expansion of the burned gases to atmospheric pressure is represented by the reverse curve from the point $p_e$ to the extreme "toe" of the diagram, and although the exhaust valve is open while the pressure is falling from the release pressure $p_e$ to atmospheric pressure, the expanding gases do some work on the piston. If it were practicable to have an exhaust-valve port large enough to let the gases drop instantaneously to atmospheric pressure when it was opened, the expansion curve could be continued to the end of the stroke, as indicated by the dotted extension, but this is, of course, impossible.

The line *e*, at practically atmospheric pressure, is traced during the exhaust or expulsion stroke of the piston. The actual pressure is very slightly above the atmosphere, of course, owing to the resistance to the flow of the burned gases presented by the exhaust port and channel and the piping leading away from the engine, but the difference is not measurable on an indicator diagram.

### The Two-stroke Cycle

In a two-stroke-cycle engine the five events just described also occur, but they are crowded into the limits of two piston strokes instead of four; hence the name of the cycle. This type of engine, however, does not draw its charge into the working cylinder by piston suction, but receives it either from a separate pump or from a reservoir to which it has been pumped, and at a pressure, therefore, above atmospheric. There are several forms of two-stroke-cycle engine; one of the simplest is that illustrated in Fig. 4. There are no valves in the cylinder wall; the inlet and exhaust ports, indicated by the letters *I* and *E*, are covered by the piston during all of its movement except a small proportion near the outer end of the travel. The inlet port *I* is connected by the annular passage *B* and ports *C* with a chamber *D* formed in the front end of the cylinder, which is provided with a stuffing-box through which the piston rod passes. This chamber *D* is really a pump cylinder. When the piston moves from the position shown toward the rear end of the cylinder, it draws a mixture of gas and air into the chamber *D* through the valve *A*, the channel *B* and the ports *C* (there are several of these ports spaced at equal distances around the wall of the chamber *D*). At the same time mixture previously taken into the power end of the cylinder is compressed between the piston and the cylinder head, just as in the four-stroke-cycle engine.

When the piston reaches the end of the compression stroke, the compressed mixture is burned, producing a rise in pressure behind the piston, as in the four-stroke cycle, and this pressure drives the piston forward on its power stroke. The forward movement compresses the mixture just drawn into the chamber *D*, with the result that when the piston passes the inlet port *I* the mixture rushes from the chamber *D* into the power end of the cylinder. In order to permit the fresh charge to enter the cylinder when the inlet port is uncovered, the exhaust port *E* is so located that the piston uncovers it before the inlet port is uncovered. This allows the burned gases to expand almost to atmospheric pressure before the fresh charge is admitted; therefore, the pressure to which this charge is compressed in the pump chamber *D* need be only a few pounds above that of the atmosphere.

From the foregoing it will be evident that a fresh charge is drawn into

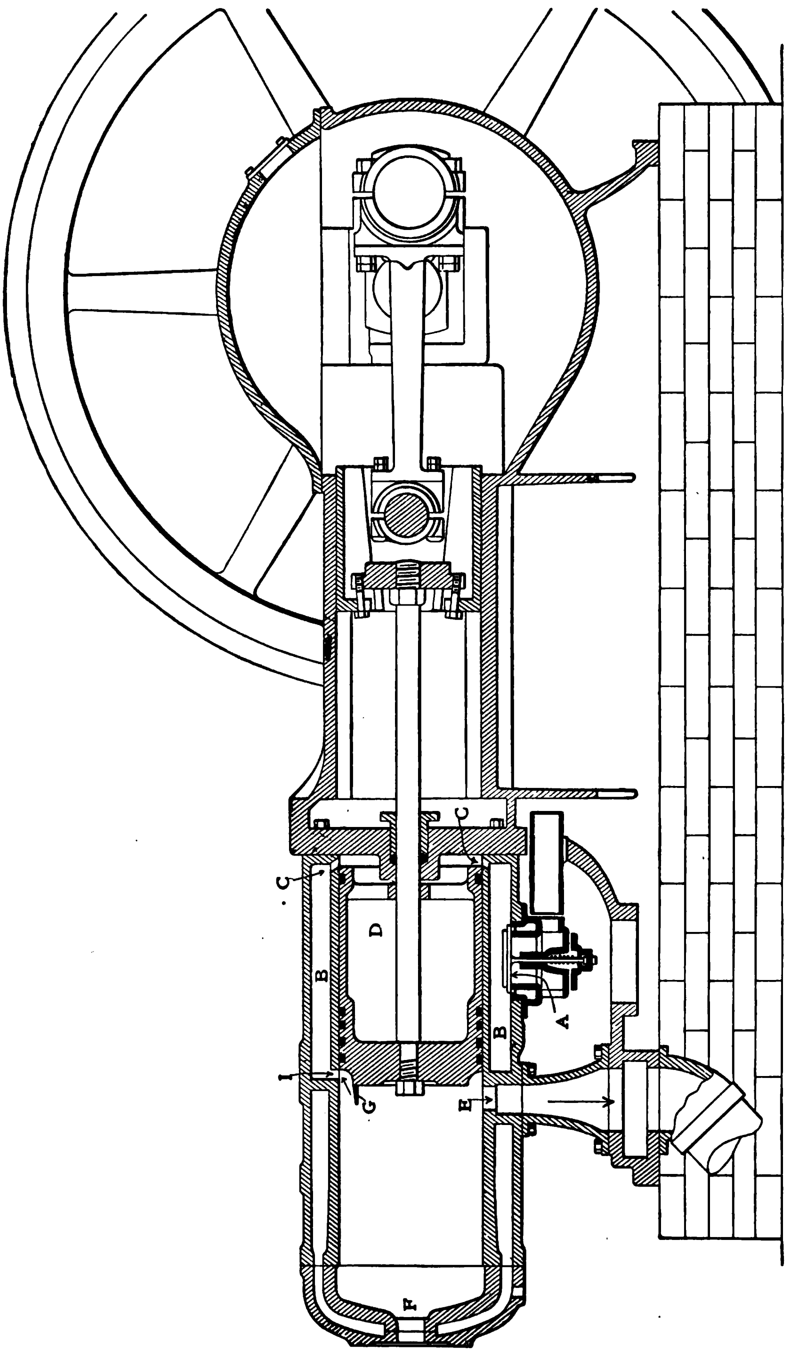



FIG. 4.—SINGLE-ACTING GAS ENGINE WORKING ON THE TWO-STROKE CYCLE.

the pump chamber *D*, and the previous charge in the cylinder is simultaneously compressed, every time the piston makes a backward stroke; that the compressed charge is fired every time the piston reaches the end of the instroke; and that the heated gases expand behind the piston and a new charge is slightly compressed in the pump chamber every time the piston makes a forward stroke, so that every forward stroke is a power stroke.

It will naturally occur to the student that while both the inlet and exhaust ports are uncovered, during the movement of the piston head from the edge of the inlet port to the end of the stroke and back again, the incoming charge, being above atmospheric pressure, will tend to pass out through the open exhaust port. This is true, and it is only by most skillful proportioning of the two ports with relation to the pump pressure and the piston speed that the escape of a considerable part of the fresh mixture with the burned gases can be avoided.

It is easily conceivable that with a certain combination of port areas, pumping pressure and piston speed, the burned gases will expand to a pressure below the pump delivery pressure, but not to atmospheric pressure, by the time the inlet port is uncovered, and that they will not get down to atmospheric pressure until the inlet port has been covered again by the piston on its return stroke. Under such conditions, the burned gases would form a sort of barrier between the fresh mixture and the exhaust port. Of course, there is some mixing of the fresh charge with the burned gases, but the incoming mixture is deflected toward the cylinder head by the baffle *G* on the piston head, so that, under the ideal combination of conditions just outlined, the burned gases with which the fresh charge mixes will be retained in the cylinder after the exhaust port is covered.

It will be clear, however, that unless proper relations are obtained between the port areas, moments of opening, pump delivery pressure and piston speed, either a good deal of the fresh mixture will escape with the exhaust gases or else too large a proportion of the burned gases will be trapped in the cylinder and reduce unnecessarily the quantity of fresh mixture that can be taken in.

Two indicator diagrams are necessary to show all of the phases through which the working medium passes during each cycle; one for the power end of the cylinder and another for the pump end, or the separate pump if one be used. Fig. 5 is a power diagram from a two-stroke-cycle engine of the type just described, and Fig. 6 is the corresponding pump diagram. The power diagram corresponds to the large loop of the four-stroke diagram, and the pump diagram corresponds roughly to the small "negative" loop of the four-stroke diagram.

The right-hand end of the power diagram, Fig. 5, will probably seem a little confused until the student becomes accustomed to the fact that the exhaust and inlet ports are open simultaneously during a small part of the piston travel. The vertical dotted line *c—d* shows where the piston begins to uncover the exhaust port on its outstroke (at the point $p_e$) and has just covered it on the instroke (at the point $p_a$). The dotted line *a—b* indicates similarly the point in the piston travel when the edge of

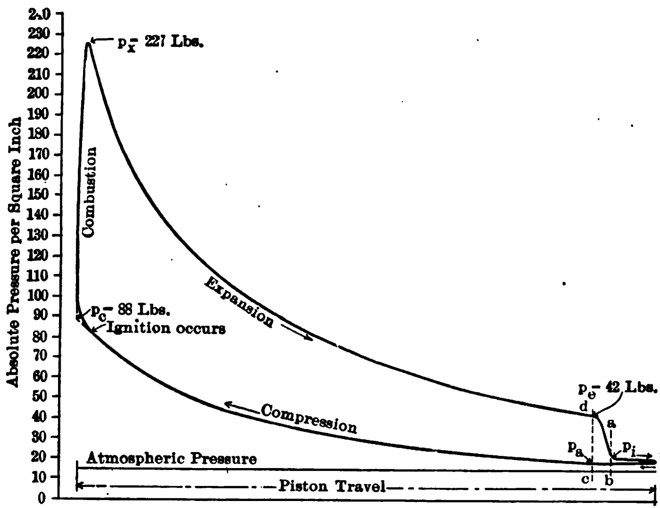



FIG. 5. — INDICATOR DIAGRAM FROM AN ILLUMINATING-GAS ENGINE WORKING ON THE TWO-STROKE CYCLE.

the piston is at the edge of the inlet port, beginning to uncover it at $p_i$ and completing its closure at the corresponding point on the lower curve.

Starting at the point $p_i$, on the diagram, the charge begins to enter the cylinder from the pump chamber and continues to enter during the remainder of the outstroke and that part of the instroke represented by the line from the toe of the diagram to the point cut by the dotted line *a—b*; the exhaust port is closed at $p_a$, and compression then begins, as indicated by the rise of the lower curve. The compression, ignition, com-

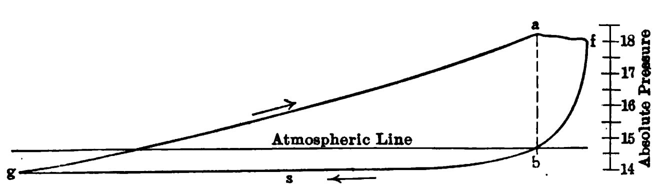



FIG. 6. — PUMP DIAGRAM, TWO-STROKE CYCLE.

bustion and expansion are the same as in the four-stroke cycle, down to the point $p_e$, where the piston begins to uncover the exhaust port. From here on, the pressure drops abruptly until the inlet port begins to open at $p_i$ and the inrush of the slightly compressed charge from the pump chamber keeps the pressure up. The burned gases continue to escape during the remainder of the outstroke and that part of the return stroke from the toe of the diagram to the point $p_a$ where the exhaust port is covered by the piston.

Referring to the pump diagram, Fig. 6, the vertical dotted line *a—b*, crossing the upper and lower curves, corresponds with the same line in Fig. 5, showing the points at which the piston covers and uncovers the inlet port of the power cylinder. Beginning at *b*, when the piston has just covered the port and stopped the escape of mixture to the cylinder, the line *s* is drawn while the piston is drawing fresh mixture into the pump chamber through the admission valve *A*, Fig. 4; this occurs simultaneously with the drawing of the compression curve of Fig. 5. Then the piston is driven forward, after the explosion in the power end of the cylinder, by the expanding gases, and compresses the mixture in the pump chamber, making the curve from *g* to *a* on the pump diagram and the expansion curve down to $p_i$ on the power diagram. At *a* the piston uncovers the inlet port of the cylinder and the rush of mixture from the pump chamber into the cylinder relieves the pump pressure, causing the drop from *a* to *f* on the diagram. As the piston returns (from right to left in Figs. 4, 5, and 6), the pressure in the pump chamber drops rapidly until the cylinder port is covered (at *b*), stopping the escape of the mixture from the pump chamber; then the drop continues more gradually until the pressure falls below that of the atmosphere, when the automatic admission valve *A*, Fig. 4, is opened by the atmospheric pressure and another charge is drawn into the pump chamber, giving the suction line of the succeeding pump diagram. It should be noted that the pressure scale in Fig. 6 is much lower than that of Figs. 5 and 3, so that the area inclosed by the loop is much larger than if the same scale had been used in all of the diagrams.

The diagrams, Figs. 3 and 5, have been slightly idealized at the "toe" in order to explain more clearly what occurs at the end of the expansion stroke in both types of engine. An accurate clean-cut "toe" cannot be obtained with the average indicator unless the speed of the engine is very slow, because the inertia of the indicator mechanism prevents the pencil from following the abrupt changes of pressure which actually occur. An ordinary diagram shows a rounded "toe" as illustrated by Figs. 25 to 30.

From the foregoing it will be evident that an engine working on the four-stroke cycle has the advantage of forcing nearly all of the burned gases out with its piston during an entire stroke and taking in a full piston displacement of fresh mixture during another entire stroke, whereas an engine working on the two-stroke cycle must take in its fresh mixture and exhaust its burned gases simultaneously and during a very small portion of two strokes.

On the other hand, an engine working on the two-stroke cycle has the advantage that one half of its piston strokes are power strokes, while the other engine gives only one power stroke in every four, the other three being devoted to intake, compression, and expulsion. Modifications of the original type of two-stroke-cycle engine have been built, moreover, in which only compressed air is admitted to the cylinder while the exhaust port is open, the fuel being delivered under pressure after the piston has closed the exhaust port. In this form of engine, it is manifestly easy to sweep the cylinder clean of burned gases and take in a cylinderful of fresh mixture without losing any of the latter. The work of air pumping, however, is a serious item in such an engine, since the air pressure must be fairly high in order to clear out the burned gases during the brief period of time available while the exhaust port is uncovered by the piston. This complete sweeping out of the burned gases is termed "scavenging," and an engine in which this is accomplished is designated a "scavenging" engine.

The difficulty of admitting a fresh charge and exhausting spent gases simultaneously and during an extremely brief interval of time, with economy, and the greater pump work required have prevented the two-stroke cycle from being as extensively applied as the four-stroke cycle.

## II

## PRESSURES AND TEMPERATURES

### Compression

In both four-stroke and two-stroke engines the pressure obtained at the end of the compression stroke is of great importance; the higher this pressure the higher will be the maximum pressure produced by combustion and the mean effective pressure of the cycle, up to a certain point. Beyond that point, which varies with differing working conditions, increasing the compression pressure does not produce any increase in the mean effective pressure, and a considerable increase will cause a decrease in mean effective pressure.

The compression pressure is determined chiefly by the relation between the volume of the clearance space (the space between the piston face and the nearest cylinder head when the crank is on the dead center) and the volume of the space "swept out" by the compression stroke of the piston. It is also influenced by the pressure which exists in the cylinder immediately before compression begins and by the loss of heat through the cylinder walls during the compression stroke; the higher the precompression pressure, the higher will be the compression pressure; the less the heat loss through the cylinder walls, the higher will be the compression pressure. These relations are simply expressed by the rule:

*Pressure before compression × Volume before compression ÷ Temperature before compression = Pressure after compression × Volume after compression ÷ Temperature after compression.*

It is even simpler, however, to express these relations in the shape of a formula, thus:

$$\frac{p_a \times V_a}{T_a} = \frac{p_c \times V_c}{T_c},$$

in which

$$\left.\begin{array}{l} p_a = \text{Absolute pressure per square inch in the cylinder,} \\ V_a = \text{Volume, in cubic feet, behind the piston,} \\ T_a = \text{Absolute temperature,} \end{array}\right\} \text{Before Compression.}$$

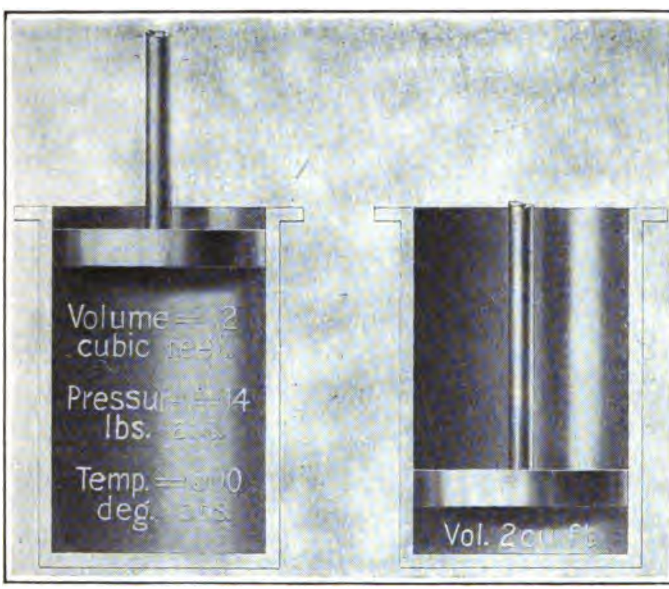
$$\left.\begin{array}{l} p_c = \text{Absolute pressure,} \\ V_c = \text{Volume, in cubic feet,} \\ T_c = \text{Absolute temperature,} \end{array}\right\} \begin{array}{c}\text{After} \\ \text{Compression.}\end{array}$$

Absolute pressure is the gauge pressure plus the atmospheric pressure, and is commonly taken at 14.7 pounds per square inch above the gauge pressure.

The volume "behind the piston" is the total space between the piston head and the cylinder head under consideration.

The absolute temperature is the Fahrenheit thermometer temperature + 460, because the absolute zero of gases is 460° below the zero of the Fahrenheit thermometer.

Reference to Fig. 7 will doubtless help the reader to grasp the meaning of the formula more readily. At $A$ is represented a cylinder and piston




A B

FIG. 7. — ILLUSTRATING COMPRESSION EFFECTS.

with a space of 12 cu. ft. between the piston and closed end of the cylinder, that is, "behind the piston." Suppose this space to be filled with air at 14 lbs. absolute pressure per square inch and 500 degrees absolute temperature (40 degrees thermometer temperature). Now suppose the piston were forced down until the space beneath it were reduced to 2 cu. ft., as indicated at $B$. If the piston and the walls of the cylinder were absolutely nonconductive to heat, so that no heat could escape, the pressure under the piston would rise to 175 lbs. absolute. Then, according to the formula:

$$\frac{14 \times 12}{500} = \frac{175 \times 2}{T_c},$$

and, consequently,

$$\frac{500}{14 \times 12} = \frac{T_c}{175 \times 2},$$

which obviously reduces to:

$$\frac{500}{168} = \frac{T_c}{350}.$$

Therefore, the absolute temperature after compression would be:

$$\frac{500}{168} \times 350 = 1041.7 \text{ degrees.}$$

Now, abandon the supposition that the piston and cylinder do not conduct heat, and assume, for example, that the escape of heat during compression was such that the temperature after compression was 900 degrees. Then the following relations would exist, according to the formula:

$$\frac{14 \times 12}{500} = \frac{p_c \times 2}{900},$$

which transposes into

$$\frac{14 \times 12}{500} \times \frac{900}{2} = p_c$$

The compression pressure would, therefore, be 151.2 lbs. per square inch instead of 175.

It is customary to consider the "compression ratio" in gas-engine calculations, rather than the actual volumes before and after compression. These volumes must be known, however, in order to ascertain the compression ratio because the ratio is merely the larger volume divided by the smaller, thus:

$$\frac{\textit{Volume before compression}}{\textit{Volume after compression}} = \textit{Compression ratio.}$$

But the ratio is more convenient to use in calculations than the separate volumes. The compression ratio in the example just given was:

$$\frac{12}{2} = 6.$$

The rise of pressure due to compression may be computed without including the temperatures, if one knows approximately the extent to which loss of heat affects the process. The formula for pressure is:

$$p_c = p_a \times r_c{}^n,$$

in which $r_c$ represents the compression ratio and the exponent $n$ depends upon the loss of heat through the cylinder walls; in practice $n$ ranges from 1.2 to 1.38, these being extreme cases. The values of $n$ most commonly obtained are from 1.28 to 1.35, a very common value being 1.30. The greater the loss of heat through the cylinder walls, the smaller will be the exponent $n$; with no loss at all it would be 1.408 for air alone and for most mixtures of air and gas used in engines.

The pressure before compression in a four-stroke-cycle engine, as explained in connection with the indicator diagram, depends on the resistance offered by the intake passages to the flow of the charge into the cylinder; it is usually 13 to 13½ lbs. absolute, but in some engines is as low as 12 lbs. and in others as high as 14 lbs.

In a two-stroke-cycle engine the pressure immediately before compression begins (when the piston has just covered the exhaust port) is usually from 14 to 16 lbs. absolute, according to the design of the ports and the delivery pressure of the pump chamber.

The increased temperature due to compression may be computed by means of a formula very similar to that for the compression pressure:

$$T_c = T_a \times r_c^{\,n-1},$$

in which the symbols have the same meaning as before. The temperature immediately before compression begins is not the temperature at which the mixture enters the valve cage, because it is heated by the hot cylinder walls and the edges of the inlet port as it enters. Ordinarily the precompression temperature is somewhere between 600 and 750 degrees absolute, very common temperatures falling between 660 and 700 degrees.

Table 1 gives compression pressures resulting from precompression pressures of 13, 13½, 14, 15 and 16 lbs., absolute, and Table 2 gives compression temperatures resulting from precompression temperatures of 620, 660, 700, 740, and 780 degrees, both with compression ratios of 3 to 8 and compression exponents of 1.3 and 1.34. Unfortunately it is impossible to predict what the compression exponent will be unless one has some knowledge of the working of an engine under the same conditions as those applying to the problem in hand, because the heat loss during compression cannot be foretold. Past practice, however, has made it certain that the heat loss can be controlled sufficiently to keep the value of the exponent somewhere near 1.3 in a well-cooled engine and 1.34 in one moderately cooled; hence, these values were used in computing the tables.

## Combustion

The rise of pressure produced by the combustion of the charge is even more difficult to estimate beforehand than the exponent of the compression formulas. The relation between the pressures and temperatures before and after combustion is the same as that already stated for compression, thus:

$$\frac{p_x \times V_x}{T_x} = \frac{p_c \times V_c}{T_c},$$

and if combustion of the entire mass of cylinder contents occurred instantaneously, while the piston was at the end of its stroke and therefore stationary, the volume after combustion ($V_x$) would be the same as the volume before combustion ($V_c$) and would not need to be considered; the formula would then reduce to

$$\frac{p_x}{T_x} = \frac{p_c}{T_c}.$$

and this transposes into

$$p_x = p_c \times \frac{T_x}{T_c}.$$

The symbols used in these formulas have the following meanings:

$p_x$ = Maximum absolute pressure per square inch after ignition.
$p_c$ = Absolute pressure per square inch due to compression.
$T_x$ = Maximum absolute temperature after ignition.
$T_c$ = Absolute temperature due to compression.

The maximum temperature after ignition is, of course, equal to the temperature due to compression plus the increase in temperature due to combustion. But this increase in temperature cannot be predicted with certainty because combustion is not instantaneous throughout the entire mass of the cylinder contents and a great deal of heat escapes from the burning gases through the cylinder walls. It should be kept in mind that the cylinder contents at the time of combustion consist of the fresh mixture taken in during the suction stroke, together with what burned gases remained in the cylinder after the previous explosion and exhaust stroke. These burned gases, having no heat value, reduce the inflammability of the total cylinder contents and make combustion less rapid than it would be if only the fresh mixture were in the cylinder.

In general, the rise of temperature produced by combustion is from 0.4 to 0.7 of what it would be with instantaneous combustion and no heat loss to the cylinder walls, so that the maximum temperature and pressure can be estimated within these limits, but this gives merely the roughest approximation. In estimating the probable output of an engine, one may assume that the rise of temperature due to combustion will be one half of the calculated value based on the unattainable perfect conditions mentioned, and thereby come within probably 20 per cent. of the truth. This question is more fully discussed in the chapter devoted to the mathematics of the gas-engine cycle, and need not enter into the present elementary considerations. It is probably sufficient now to point out that the maximum temperature after combustion may be as high as 3,000 degrees absolute, and the maximum pressure as high as 400 lbs. per square inch absolute; these are high figures, however, the more usual values being about 2,300 degrees absolute temperature and 250 lbs. absolute pressure.

**Table 3-A. Average Pressure Rise per Square Inch Produced by Combustion**

| Comp. ratio $r_c$ | Illuminating Gas B.t.u. per cubic foot, cold 600 | 625 | 650 | Gasolene | Kerosene |
|---|---|---|---|---|---|
| 4.0 | 135 | 140 | 146 | 195 | 168 |
| 4.1 | 139 | 145 | 151 | 201 | 174 |
| 4.2 | 144 | 150 | 156 | 208 | 179 |
| 4.3 | 148 | 155 | 161 | 214 | 185 |
| 4.4 | 153 | 159 | 166 | 221 | 190 |
| 4.5 | 157 | 164 | 170 | 227 | 196 |
| 4.6 | 162 | 169 | 175 | 234 | 202 |
| 4.7 | 166 | 173 | 180 | 240 | 207 |
| 4.8 | 171 | 178 | 185 | 247 | 213 |
| 4.9 | 175 | 183 | 190 | 253 | 218 |
| 5.0 | 180 | 187 | 195 | 260 | 224 |

**Table 3-B. Average Pressure Rise per Square Inch Produced by Combustion**

| Comp. ratio $r_c$ | Natural Gas B.t.u. per cubic foot, cold 900 | 950 | 1000 | 1050 | 1100 |
|---|---|---|---|---|---|
| 5.0 | 173 | 182 | 192 | 202 | 211 |
| 5.1 | 177 | 187 | 197 | 207 | 216 |
| 5.2 | 181 | 191 | 202 | 212 | 222 |
| 5.3 | 186 | 196 | 206 | 217 | 227 |
| 5.4 | 190 | 200 | 211 | 222 | 232 |
| 5.5 | 194 | 205 | 216 | 227 | 238 |
| 5.6 | 199 | 210 | 221 | 232 | 243 |
| 5.7 | 203 | 214 | 226 | 237 | 248 |
| 5.8 | 207 | 219 | 230 | 242 | 253 |
| 5.9 | 212 | 223 | 235 | 247 | 259 |
| 6.0 | 216 | 228 | 240 | 252 | 264 |

**Table 3-C. Average Pressure Rise per Square Inch Produced by Combustion**

| Comp. ratio $r_c$ | Producer Gas B.t.u. per cubic foot, cold 120 | 130 | 140 | 150 | 160 |
|---|---|---|---|---|---|
| 6.0 | 180 | 195 | 210 | 225 | 240 |
| 6.1 | 184 | 199 | 214 | 229 | 245 |
| 6.2 | 187 | 203 | 218 | 234 | 250 |
| 6.3 | 191 | 207 | 223 | 238 | 254 |
| 6.4 | 194 | 210 | 227 | 243 | 259 |
| 6.5 | 198 | 214 | 231 | 247 | 264 |
| 6.6 | 201 | 218 | 235 | 252 | 269 |
| 6.7 | 205 | 222 | 239 | 256 | 273 |
| 6.8 | 209 | 226 | 243 | 261 | 278 |
| 6.9 | 212 | 230 | 248 | 265 | 283 |
| 7.0 | 216 | 234 | 252 | 270 | 288 |

**Table 3-D. Average Pressure Rise per Square Inch Produced by Combustion**

| Comp. ratio $r_c$ | Blast Furnace Gas B.t.u. per cubic foot, cold 80 | 85 | 90 | 95 | 100 |
|---|---|---|---|---|---|
| 7.0 | 169 | 180 | 190 | 201 | 211 |
| 7.1 | 172 | 183 | 193 | 204 | 215 |
| 7.2 | 175 | 186 | 196 | 207 | 218 |
| 7.3 | 178 | 189 | 200 | 211 | 222 |
| 7.4 | 180 | 192 | 203 | 214 | 225 |
| 7.5 | 183 | 194 | 206 | 217 | 229 |
| 7.6 | 186 | 197 | 209 | 220 | 232 |
| 7.7 | 189 | 200 | 212 | 224 | 236 |
| 7.8 | 192 | 203 | 215 | 227 | 239 |
| 7.9 | 194 | 206 | 218 | 230 | 243 |
| 8.0 | 197 | 209 | 222 | 234 | 246 |

Tables 3-A to 3-D give the average rises of pressure produced by the combustion of different fuels, as obtained in practice.

## Expansion and Exhaust

During the power stroke of the piston the burned and burning gases expand according to exactly the same law that they followed during compression. If there were no gain or loss of heat by the gases during expansion, the pressure and temperature would fall at exactly the same rates at which they would have risen during compression with no heat loss or gain; consequently, if the gases could receive heat from any source during expansion at the same rate at which heat was lost to the cylinder walls during compression, the rates at which the pressure and temperature would change would be the same during expansion as during compression. In other words, if the gases could receive heat during expansion at the same rate that heat was lost during compression, the pressure at any point of the expansion curve divided by the pressure at the point on the compression curve vertically below it would be the same at all points of the two curves. Thus, if the expansion pressure were three times the compression pressure at a point one tenth of the piston travel from the left-hand end of the diagram, it would be three times the compression pressure at three tenths, three fourths, or any other point of the travel.

In practice, however, this result is rarely obtained. The gases do receive heat during expansion, much of it from continued burning after the piston has commenced its power stroke, because combustion was not instantaneous. In addition to this, as the temperature of the expanding gases falls some heat is given back to them by the cylinder walls, where it was stored during combustion. The drop in the expansion curve, however, is usually less rapid than the rise in the compression curve, but it occasionally happens that it is the other way, the gases receiving less heat by after-burning and restoration from the cylinder walls during expansion than they lost during compression. Fig. 8 is an example of this kind. At the moment of maximum pressure, that pressure was 2.9 times the compression pressure that existed when the piston was in the same position during the compression stroke (not the same point of the compression stroke, but the same actual position in the cylinder). When the expansion pressure had fallen to 173 lbs., it was 2.88 times the pressure on the compression curve corresponding to the same piston position in the cylinder; this continued true for some distance along the two curves, but at 114 lbs. on the expansion curve the ratio of expansion to compression pressures had fallen to 2.85, and at 85 lbs. it went down to 2.83. Beyond this point the gases received heat more rapidly, as shown by the flattening of the expansion curve, and at the moment of release the pressure ratio rose to 2.9 again. From this illustration it will be evident that calculations relating to pressure variations under the conditions of gas-engine operation cannot be made beforehand with any approach to strict accuracy.

The expansion ratio is never the same as the compression ratio because the exhaust port is opened before the end of the expansion stroke. The difference between the two ratios is greater in a four-stroke-cycle engine than in a two-stroke, because in the former the compression begins at the very beginning of the compression stroke, while in the latter it does not begin until the piston has covered the exhaust port. In the four-stroke cycle, the expansion ratio is always smaller than the compression ratio, but in the two-stroke cycle as ordinarily carried out it is a trifle larger. Reference to Figs. 9 and 10 will show why this is so. In Fig. 9 is shown

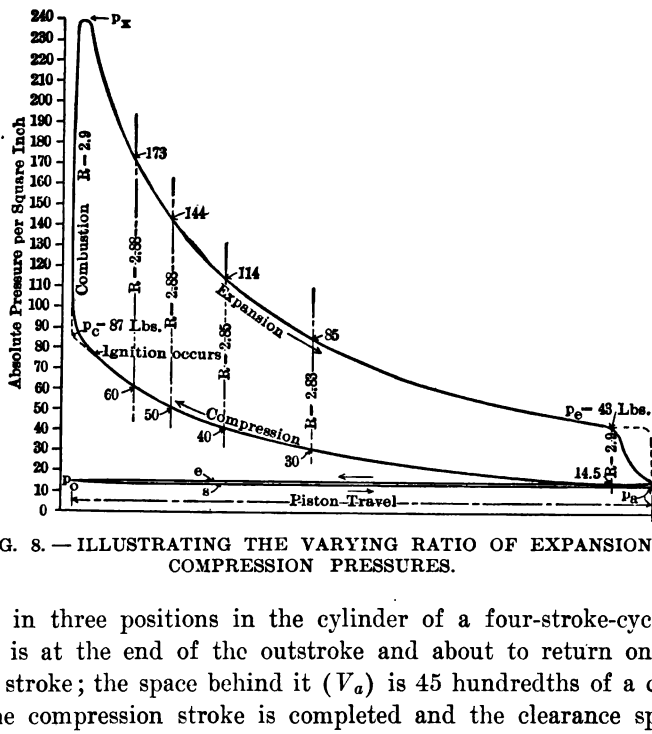



FIG. 8. — ILLUSTRATING THE VARYING RATIO OF EXPANSION TO COMPRESSION PRESSURES.

a piston in three positions in the cylinder of a four-stroke-cycle engine. At $A$ it is at the end of the outstroke and about to return on the compression stroke; the space behind it ($V_a$) is 45 hundredths of a cubic foot. At $B$ the compression stroke is completed and the clearance space ($V_c$) is one tenth of a cubic foot. The compression ratio is therefore:

$$0.45 \div 0.1 = 4\tfrac{1}{2}.$$

At $C$ the piston has reached the position in the expansion stroke where the exhaust valve (not shown) opens, and the space behind it ($V_e$) is 41 hundredths of a cubic foot; this position is (in this case) at 87 per cent. of the expansion stroke, the extreme end of which is indicated by the dotted line. Now if the expanding gases dropped to atmospheric pressure the instant the exhaust valve opened, the expansion ratio would be:

$$0.41 \div 0.1 = 4.1$$

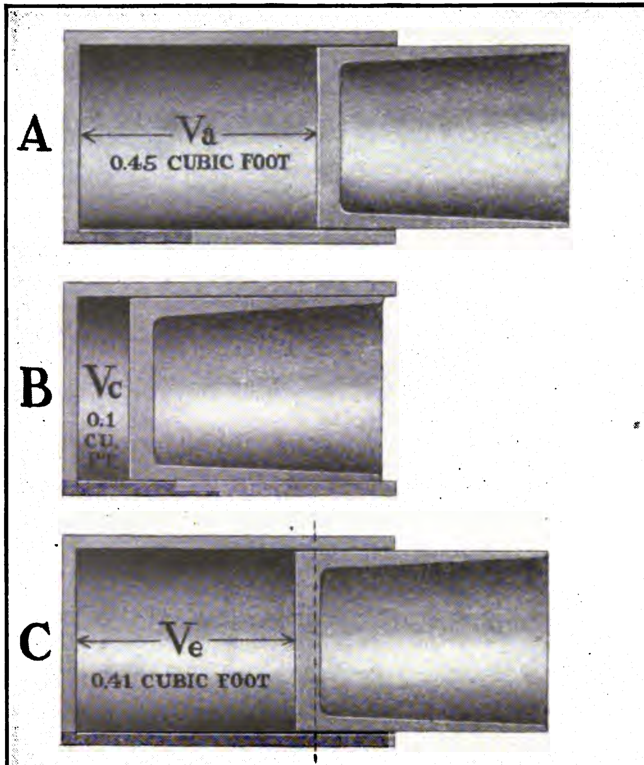



FIG. 9. — ILLUSTRATING DIFFERENCE BETWEEN COMPRESSION AND EXPANSION RATIOS IN THE FOUR-STROKE CYCLE.

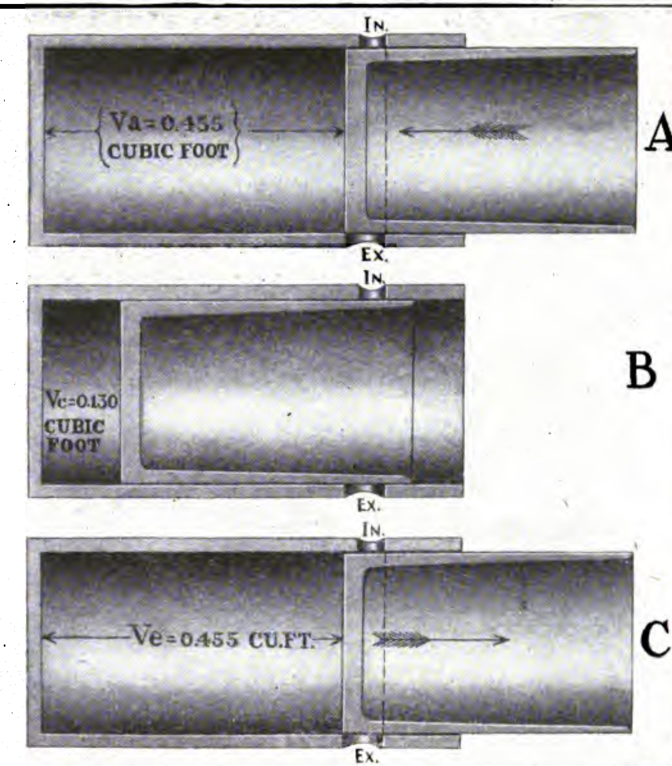



FIG. 10. — ILLUSTRATING SIMILARITY BETWEEN COMPRESSION AND EXPANSION RATIOS IN THE TWO-STROKE CYCLE.

as compared with $4\frac{1}{2}$ for the compression ratio. The gases do not drop instantly to atmospheric pressure, however. Usually the drop is sufficiently gradual to make the mean effective pressure the same that it would be if the exhaust valve did not open until the piston traveled nearly one half of the remainder of the stroke before the exhaust valve opened, and the pressure then dropped instantly to atmospheric. Even with this allowance, the expansion ratio cannot be as great as the compression ratio, because the effective part of the expansion stroke is still slightly less than the full piston stroke.

Fig. 10 illustrates the two-stroke conditions. At $A$ the piston, moving in the direction of the arrow, has just covered the exhaust port and begun compression; the space behind it is 455 thousandths of a cubic foot. At $B$ the compression stroke is completed and the clearance space is 130 thousandths of a cubic foot; the compression ratio, therefore, is

$$\frac{455}{130} = 3\tfrac{1}{2}.$$

At $C$ the piston, moving outward, is about to uncover the exhaust port, and is in practically the same position as when compression began, at $A$. If the expanding gases dropped instantly to atmospheric pressure when the exhaust port began to open, the expansion ratio would be practically the same as the compression ratio, but the drop is not instantaneous, and the gradual fall of pressure amounts to an extension of the effective part of the power stroke, making the expansion ratio greater than the compression ratio. The dotted lines indicate the end of the outstroke and the beginning of the instroke.

The pressure at which the expanding gases are released bears the same general relation to the maximum pressure of combustion that the precompression pressure does to the compression pressure. Thus:

$$p_e = p_x \div r_e^n,$$

$r_e$ being the expansion ratio, and $n$ the exponent of the expansion curve.

The temperature of the gases at the moment of release is

$$T_e = T_x \div r_e^{n-1}.$$

By transposing these formulas their similarity to the compression formulas will be more apparent, thus:

$$p_e \times r_e^n = p_x$$

and

$$T_e \times r_e^{n-1} = T_x.$$

Since the release pressure and temperature are determined by the explosion pressure and temperature, however, the first form is the correct one for both formulas. Table 4 gives release pressures and Table 5 gives

# Table 4. Absolute Pressures per Square Inch at Release

CORRESPONDING TO EXPLOSION PRESSURES COMMONLY OBTAINED.

Note:—The expansion ratios in the left-hand column are based on the volume behind the piston when the exhaust valve begins to open.

| Expansion Ratio $r_e$ | $n_e = 1.29$ | | | | | $n_e = 1.32$ | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | $p_x =$ 240 | $p_x =$ 270 | $p_x =$ 300 | $p_x =$ 330 | $p_x =$ 360 | $p_x =$ 240 | $p_x =$ 270 | $p_x =$ 300 | $p_x =$ 330 | $p_x =$ 360 |
| 3.00 | 58.2 | 65.4 | 72.7 | 80.0 | 87.2 | 56.3 | 63.3 | 70.4 | 77.4 | 84.4 |
| 3.05 | 56.9 | 64.0 | 71.2 | 78.3 | 85.4 | 55.1 | 62.0 | 68.8 | 75.7 | 82.6 |
| 3.10 | 55.8 | 62.7 | 69.7 | 76.7 | 83.6 | 53.9 | 60.6 | 67.4 | 74.1 | 80.9 |
| 3.15 | 54.6 | 61.4 | 68.3 | 75.1 | 81.9 | 52.8 | 59.4 | 66.0 | 72.6 | 79.2 |
| 3.20 | 53.5 | 60.2 | 66.9 | 73.6 | 80.3 | 51.7 | 58.2 | 64.6 | 71.1 | 77.5 |
| 3.25 | 52.5 | 59.0 | 65.6 | 72.1 | 78.7 | 50.6 | 57.0 | 63.3 | 69.6 | 76.0 |
| 3.30 | 51.4 | 57.9 | 64.3 | 70.7 | 77.2 | 49.6 | 55.8 | 62.0 | 68.3 | 74.5 |
| 3.35 | 50.5 | 56.8 | 63.1 | 69.4 | 75.7 | 48.7 | 54.7 | 60.8 | 66.9 | 73.0 |
| 3.40 | 49.5 | 55.7 | 61.9 | 68.1 | 74.2 | 47.7 | 53.7 | 59.6 | 65.6 | 71.6 |
| 3.45 | 48.6 | 54.6 | 60.7 | 66.8 | 72.9 | 46.8 | 52.7 | 58.5 | 64.4 | 70.2 |
| 3.50 | 47.7 | 53.6 | 59.6 | 65.6 | 71.5 | 45.9 | 51.7 | 57.4 | 63.1 | 68.9 |
| 3.55 | 46.8 | 52.7 | 58.5 | 64.4 | 70.2 | 45.1 | 50.7 | 56.3 | 62.0 | 67.6 |
| 3.60 | 46.0 | 51.7 | 57.5 | 63.2 | 68.9 | 44.2 | 49.8 | 55.3 | 60.8 | 66.4 |
| 3.65 | 45.2 | 50.8 | 56.5 | 62.1 | 67.8 | 43.4 | 48.9 | 54.3 | 59.7 | 65.2 |
| 3.70 | 44.4 | 49.9 | 55.5 | 61.0 | 66.6 | 42.7 | 48.0 | 53.3 | 58.7 | 64.0 |
| 3.75 | 43.6 | 49.1 | 54.5 | 60.0 | 65.4 | 41.9 | 47.2 | 52.4 | 57.7 | 62.9 |
| 3.80 | 42.9 | 48.2 | 53.6 | 58.9 | 64.3 | 41.2 | 46.4 | 51.5 | 56.7 | 61.8 |
| 3.85 | 42.2 | 47.4 | 52.7 | 58.0 | 63.2 | 40.5 | 45.6 | 50.6 | 55.7 | 60.7 |
| 3.90 | 41.5 | 46.7 | 51.8 | 57.0 | 62.2 | 39.8 | 44.8 | 49.8 | 54.7 | 59.7 |
| 3.95 | 40.8 | 45.9 | 51.0 | 56.1 | 61.2 | 39.1 | 44.0 | 48.9 | 53.8 | 58.7 |
| 4.00 | 40.1 | 45.2 | 50.2 | 55.2 | 60.2 | 38.5 | 43.3 | 48.1 | 52.9 | 57.8 |
| 4.05 | 39.5 | 44.4 | 49.4 | 54.3 | 59.2 | 37.9 | 42.6 | 47.3 | 52.1 | 56.8 |
| 4.10 | 38.9 | 43.7 | 48.6 | 53.5 | 58.3 | 37.3 | 41.9 | 46.6 | 51.2 | 55.9 |
| 4.15 | 38.3 | 43.1 | 47.8 | 52.6 | 57.4 | 36.7 | 41.3 | 45.8 | 50.4 | 55.0 |
| 4.20 | 37.7 | 42.4 | 47.1 | 51.8 | 56.5 | 36.1 | 40.6 | 45.1 | 49.6 | 54.2 |
| 4.25 | 37.1 | 41.8 | 46.4 | 51.0 | 55.7 | 35.5 | 40.0 | 44.4 | 48.9 | 53.3 |
| 4.30 | 36.6 | 41.1 | 45.7 | 50.3 | 54.8 | 35.0 | 39.4 | 43.7 | 48.1 | 52.5 |
| 4.35 | 36.0 | 40.5 | 45.0 | 49.5 | 54.0 | 34.5 | 38.8 | 43.1 | 47.4 | 51.7 |
| 4.40 | 35.5 | 39.9 | 44.4 | 48.8 | 53.2 | 34.0 | 38.2 | 42.4 | 46.7 | 50.9 |
| 4.45 | 35.0 | 39.4 | 43.7 | 48.1 | 52.5 | 33.4 | 37.6 | 41.8 | 46.0 | 50.2 |
| 4.50 | 34.5 | 38.8 | 43.1 | 47.4 | 51.7 | 33.0 | 37.1 | 41.2 | 45.3 | 49.4 |
| 4.55 | 34.0 | 38.2 | 42.5 | 46.7 | 51.0 | 32.5 | 36.5 | 40.6 | 44.7 | 48.7 |
| 4.60 | 33.5 | 37.7 | 41.9 | 46.1 | 50.3 | 32.0 | 36.0 | 40.0 | 44.0 | 48.0 |
| 4.65 | 33.1 | 37.2 | 41.3 | 45.4 | 49.6 | 31.6 | 35.5 | 39.5 | 43.4 | 47.3 |
| 4.70 | 32.6 | 36.7 | 40.7 | 44.8 | 48.9 | 31.1 | 35.0 | 38.9 | 42.8 | 46.7 |
| 4 75 | 32.2 | 36.2 | 40.2 | 44.2 | 48.2 | 30.7 | 34.5 | 38.4 | 42.2 | 46.0 |
| 4.80 | 31.7 | 35.7 | 39.6 | 43.6 | 47.6 | 30.3 | 34.1 | 37.8 | 41.6 | 45.4 |
| 4.85 | 31.3 | 35.2 | 39.1 | 43.0 | 46.9 | 29.9 | 33.6 | 37.3 | 41.1 | 44.8 |
| 4.90 | 30.9 | 34.8 | 38.6 | 42.5 | 46.3 | 29.5 | 33.1 | 36.8 | 40.5 | 44.2 |
| 4.95 | 30.5 | 34.3 | 38.1 | 41.9 | 45.7 | 29.1 | 32.7 | 36.3 | 40.0 | 43.6 |
| 5.00 | 30.1 | 33.9 | 37.6 | 41.4 | 45.1 | 28.7 | 32.3 | 35.8 | 39.4 | 43.0 |
| 5.10 | 29.3 | 33.0 | 36.7 | 40.3 | 44.0 | 27.9 | 31.4 | 34.9 | 38.4 | 41.9 |
| 5.20 | 28.6 | 32.2 | 35.8 | 39.3 | 42.9 | 27.2 | 30.6 | 34.0 | 37.4 | 40.8 |
| 5.30 | 27.9 | 31.4 | 34.9 | 38.4 | 41.9 | 26.6 | 29.9 | 33.2 | 36.6 | 39.8 |
| 5.40 | 27.3 | 30.7 | 34.1 | 37.5 | 40.9 | 25.9 | 29.1 | 32.4 | 35.6 | 38.9 |
| 5.50 | 26.6 | 29.9 | 33.3 | 36.6 | 39.9 | 25.3 | 28.5 | 31.6 | 34.8 | 37.9 |
| 5.60 | 26.0 | 29.3 | 32.5 | 35.8 | 39.0 | 24.7 | 27.8 | 30.9 | 34.0 | 37.0 |
| 5.70 | 25.4 | 28.6 | 31.8 | 34.9 | 38.1 | 24.1 | 27.1 | 30.2 | 33.2 | 36.2 |
| 5.80 | 24.9 | 28.0 | 31.1 | 34.2 | 37.3 | 23.6 | 26.6 | 29.5 | 32.4 | 35.4 |
| 5.90 | 24.3 | 27.3 | 30.4 | 33.4 | 36.5 | 23.0 | 25.9 | 28.8 | 31.7 | 34.6 |
| 6.00 | 23.8 | 26.8 | 29.7 | 32.7 | 35.7 | 22.5 | 25.4 | 28.2 | 31.0 | 33.8 |
| 6.10 | 23.3 | 26.2 | 29.1 | 32.0 | 34.9 | 22.1 | 24.8 | 27.6 | 30.3 | 33.1 |
| 6.20 | 22.8 | 25.7 | 28.5 | 31.4 | 34.2 | 21.6 | 24.3 | 27.0 | 29.7 | 32.4 |
| 6.30 | 22.3 | 25.1 | 27.9 | 30.7 | 33.5 | 21.1 | 23.8 | 26.4 | 29.1 | 31.7 |
| 6.40 | 21.9 | 24.6 | 27.4 | 30.1 | 32.8 | 20.7 | 23.3 | 25.9 | 28.5 | 31.1 |
| 6.50 | 21.5 | 24.1 | 26.8 | 29.5 | 32.2 | 20.3 | 22.8 | 25.4 | 27.9 | 30.4 |
| 6.60 | 21.0 | 23.7 | 26.3 | 28.9 | 31.6 | 19.9 | 22.4 | 24.8 | 27.3 | 29.8 |
| 6.70 | 20.6 | 23.2 | 25.8 | 28.4 | 30.9 | 19.5 | 21.9 | 24.4 | 26.8 | 29.2 |
| 6.80 | 20.2 | 22.8 | 25.3 | 27.8 | 30.4 | 19.1 | 21.5 | 23.9 | 26.3 | 28.7 |
| 6.90 | 19.9 | 22.3 | 24.8 | 27.3 | 29.8 | 18.7 | 21.1 | 23.4 | 25.8 | 28.1 |
| 7.00 | 19.5 | 21.9 | 24.4 | 26.8 | 29.2 | 18.4 | 20.7 | 23.0 | 25.3 | 27.6 |
| 7.10 | 19.1 | 21.5 | 23.9 | 26.3 | 28.7 | 18.1 | 20.3 | 22.6 | 24.8 | 27.1 |
| 7.20 | 18.8 | 21.2 | 23.5 | 25.9 | 28.2 | 17.7 | 19.9 | 22.2 | 24.4 | 26.6 |
| 7.30 | 18.5 | 20.8 | 23.1 | 25.4 | 27.7 | 17.4 | 19.6 | 21.8 | 23.9 | 26.1 |
| 7.40 | 18.2 | 20.4 | 22.7 | 25.0 | 27.2 | 17.1 | 19.2 | 21.4 | 23.5 | 25.6 |
| 7.50 | 17.8 | 20.1 | 22.3 | 24.5 | 26.8 | 16.8 | 18.9 | 21.0 | 23.1 | 25.2 |
| 7.60 | 17.5 | 19.7 | 21.9 | 24.1 | 26.3 | 16.5 | 18.6 | 20.6 | 22.7 | 24.8 |
| 7.70 | 17.2 | 19.4 | 21.6 | 23.7 | 25.9 | 16.2 | 18.2 | 20.3 | 22.3 | 24.3 |
| 7.80 | 17.0 | 19.1 | 21.2 | 23.3 | 25.4 | 15.9 | 17.9 | 19.9 | 21.9 | 23.9 |
| 7.90 | 16.7 | 18.8 | 20.9 | 22.9 | 25.0 | 15.7 | 17.6 | 19.6 | 21.6 | 23.5 |
| 8.00 | 16.4 | 18.5 | 20.5 | 22.6 | 24.6 | 15.4 | 17.3 | 19.3 | 21.2 | 23.1 |

# Table 5. Absolute Temperatures at Release

CORRESPONDING TO EXPLOSION TEMPERATURES COMMONLY OBTAINED.

Note:—The expansion ratio is based on the volume behind the piston when the exhaust valve begins to open.

| Expansion Ratio $r_e$ | $n_e = 1.29$ | | | | | $n_e = 1.32$ | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | $T_x = 1800$ | $T_x = 2100$ | $T_x = 2400$ | $T_x = 2700$ | $T_x = 3000$ | $T_x = 1800$ | $T_x = 2100$ | $T_x = 2400$ | $T_x = 2700$ | $T_x = 3000$ |
| 3.00 | 1309 | 1527 | 1745 | 1963 | 2182 | 1266 | 1478 | 1689 | 1900 | 2111 |
| 3.05 | 1303 | 1520 | 1737 | 1954 | 2171 | 1260 | 1470 | 1680 | 1890 | 2100 |
| 3.10 | 1297 | 1513 | 1729 | 1945 | 2161 | 1253 | 1462 | 1671 | 1880 | 2089 |
| 3.15 | 1290 | 1505 | 1721 | 1936 | 2151 | 1247 | 1455 | 1662 | 1870 | 2078 |
| 3.20 | 1285 | 1499 | 1713 | 1927 | 2141 | 1241 | 1447 | 1654 | 1861 | 2068 |
| 3.25 | 1279 | 1492 | 1705 | 1918 | 2131 | 1234 | 1440 | 1646 | 1852 | 2057 |
| 3.30 | 1273 | 1485 | 1698 | 1910 | 2122 | 1228 | 1433 | 1638 | 1843 | 2047 |
| 3.35 | 1268 | 1479 | 1690 | 1902 | 2113 | 1223 | 1426 | 1630 | 1834 | 2038 |
| 3.40 | 1262 | 1473 | 1683 | 1894 | 2104 | 1217 | 1420 | 1622 | 1825 | 2028 |
| 3.45 | 1257 | 1466 | 1676 | 1885 | 2095 | 1211 | 1413 | 1615 | 1817 | 2018 |
| 3.50 | 1252 | 1460 | 1669 | 1878 | 2086 | 1205 | 1406 | 1607 | 1808 | 2009 |
| 3.55 | 1246 | 1454 | 1662 | 1871 | 2077 | 1200 | 1400 | 1600 | 1800 | 2000 |
| 3.60 | 1241 | 1448 | 1655 | 1862 | 2069 | 1195 | 1394 | 1593 | 1792 | 1991 |
| 3.65 | 1237 | 1443 | 1649 | 1855 | 2061 | 1189 | 1388 | 1586 | 1784 | 1982 |
| 3.70 | 1232 | 1437 | 1642 | 1848 | 2053 | 1184 | 1382 | 1579 | 1776 | 1974 |
| 3.75 | 1227 | 1431 | 1636 | 1840 | 2045 | 1179 | 1376 | 1572 | 1769 | 1965 |
| 3.80 | 1222 | 1426 | 1630 | 1833 | 2037 | 1174 | 1370 | 1566 | 1761 | 1957 |
| 3.85 | 1218 | 1420 | 1623 | 1826 | 2029 | 1169 | 1364 | 1559 | 1754 | 1949 |
| 3.90 | 1213 | 1415 | 1617 | 1819 | 2022 | 1164 | 1359 | 1553 | 1747 | 1941 |
| 3.95 | 1209 | 1410 | 1611 | 1813 | 2014 | 1160 | 1353 | 1546 | 1740 | 1933 |
| 4.00 | 1204 | 1405 | 1606 | 1806 | 2007 | 1155 | 1348 | 1540 | 1733 | 1925 |
| 4.05 | 1200 | 1400 | 1600 | 1800 | 2000 | 1151 | 1342 | 1534 | 1726 | 1918 |
| 4.10 | 1196 | 1395 | 1594 | 1793 | 1993 | 1146 | 1337 | 1528 | 1719 | 1910 |
| 4.15 | 1191 | 1390 | 1589 | 1787 | 1986 | 1142 | 1332 | 1522 | 1712 | 1903 |
| 4.20 | 1187 | 1385 | 1583 | 1781 | 1978 | 1137 | 1327 | 1516 | 1706 | 1895 |
| 4.25 | 1183 | 1380 | 1578 | 1775 | 1972 | 1133 | 1322 | 1510 | 1699 | 1888 |
| 4.30 | 1179 | 1376 | 1572 | 1769 | 1965 | 1129 | 1317 | 1505 | 1693 | 1881 |
| 4.35 | 1175 | 1371 | 1567 | 1763 | 1959 | 1124 | 1312 | 1499 | 1687 | 1874 |
| 4.40 | 1171 | 1366 | 1562 | 1757 | 1952 | 1120 | 1307 | 1494 | 1681 | 1867 |
| 4.45 | 1167 | 1362 | 1557 | 1751 | 1946 | 1116 | 1302 | 1488 | 1675 | 1861 |
| 4.50 | 1164 | 1358 | 1552 | 1746 | 1940 | 1112 | 1298 | 1483 | 1668 | 1854 |
| 4.55 | 1160 | 1353 | 1547 | 1740 | 1933 | 1108 | 1293 | 1478 | 1663 | 1847 |
| 4.60 | 1156 | 1349 | 1542 | 1734 | 1927 | 1105 | 1289 | 1473 | 1657 | 1841 |
| 4.65 | 1153 | 1345 | 1537 | 1729 | 1921 | 1101 | 1284 | 1468 | 1651 | 1835 |
| 4.70 | 1149 | 1341 | 1532 | 1724 | 1915 | 1097 | 1280 | 1463 | 1645 | 1828 |
| 4.75 | 1146 | 1337 | 1528 | 1719 | 1910 | 1093 | 1275 | 1458 | 1640 | 1822 |
| 4.80 | 1142 | 1332 | 1523 | 1713 | 1904 | 1090 | 1271 | 1453 | 1634 | 1816 |
| 4.85 | 1139 | 1328 | 1518 | 1708 | 1898 | 1086 | 1267 | 1448 | 1629 | 1810 |
| 4.90 | 1135 | 1324 | 1514 | 1703 | 1892 | 1082 | 1263 | 1443 | 1624 | 1804 |
| 4.95 | 1132 | 1321 | 1509 | 1698 | 1887 | 1079 | 1259 | 1439 | 1618 | 1798 |
| 5.00 | 1129 | 1317 | 1505 | 1693 | 1881 | 1075 | 1255 | 1434 | 1613 | 1792 |
| 5.10 | 1122 | 1309 | 1496 | 1683 | 1870 | 1069 | 1247 | 1425 | 1603 | 1781 |
| 5.20 | 1116 | 1302 | 1488 | 1674 | 1860 | 1062 | 1239 | 1416 | 1593 | 1770 |
| 5.30 | 1110 | 1295 | 1480 | 1665 | 1850 | 1056 | 1232 | 1407 | 1583 | 1759 |
| 5.40 | 1104 | 1288 | 1472 | 1656 | 1840 | 1049 | 1224 | 1399 | 1574 | 1749 |
| 5.50 | 1098 | 1281 | 1464 | 1647 | 1830 | 1043 | 1217 | 1391 | 1565 | 1739 |
| 5.60 | 1092 | 1274 | 1456 | 1638 | 1820 | 1037 | 1210 | 1383 | 1556 | 1729 |
| 5.70 | 1087 | 1268 | 1449 | 1630 | 1811 | 1031 | 1203 | 1375 | 1547 | 1719 |
| 5.80 | 1081 | 1261 | 1441 | 1622 | 1802 | 1026 | 1197 | 1367 | 1538 | 1709 |
| 5.90 | 1076 | 1255 | 1434 | 1614 | 1793 | 1020 | 1190 | 1360 | 1530 | 1700 |
| 6.00 | 1070 | 1249 | 1427 | 1606 | 1784 | 1015 | 1184 | 1353 | 1522 | 1691 |
| 6.10 | 1065 | 1243 | 1421 | 1598 | 1776 | 1009 | 1177 | 1346 | 1514 | 1682 |
| 6.20 | 1060 | 1237 | 1414 | 1591 | 1767 | 1004 | 1171 | 1339 | 1506 | 1673 |
| 6.30 | 1056 | 1231 | 1407 | 1583 | 1759 | 999 | 1165 | 1332 | 1498 | 1665 |
| 6.40 | 1051 | 1226 | 1401 | 1576 | 1751 | 994 | 1159 | 1325 | 1491 | 1656 |
| 6.50 | 1046 | 1220 | 1395 | 1569 | 1743 | 989 | 1154 | 1318 | 1483 | 1648 |
| 6.60 | 1041 | 1215 | 1389 | 1562 | 1736 | 984 | 1148 | 1312 | 1476 | 1640 |
| 6.70 | 1037 | 1210 | 1382 | 1555 | 1728 | 979 | 1143 | 1306 | 1469 | 1632 |
| 6.80 | 1032 | 1204 | 1376 | 1549 | 1721 | 975 | 1137 | 1300 | 1462 | 1625 |
| 6.90 | 1028 | 1199 | 1371 | 1542 | 1713 | 970 | 1132 | 1294 | 1455 | 1617 |
| 7.00 | 1024 | 1194 | 1365 | 1536 | 1706 | 966 | 1127 | 1288 | 1449 | 1610 |
| 7.10 | 1020 | 1189 | 1359 | 1529 | 1699 | 961 | 1122 | 1282 | 1442 | 1602 |
| 7.20 | 1015 | 1185 | 1354 | 1523 | 1692 | 957 | 1117 | 1276 | 1436 | 1595 |
| 7.30 | 1011 | 1180 | 1348 | 1517 | 1686 | 953 | 1112 | 1270 | 1429 | 1588 |
| 7.40 | 1007 | 1175 | 1343 | 1511 | 1679 | 949 | 1107 | 1265 | 1423 | 1581 |
| 7.50 | 1003 | 1171 | 1338 | 1505 | 1672 | 945 | 1102 | 1259 | 1417 | 1574 |
| 7.60 | 1000 | 1166 | 1333 | 1499 | 1666 | 941 | 1097 | 1254 | 1411 | 1568 |
| 7.70 | 996 | 1162 | 1328 | 1494 | 1660 | 937 | 1093 | 1249 | 1405 | 1561 |
| 7.80 | 992 | 1157 | 1323 | 1488 | 1654 | 933 | 1088 | 1244 | 1399 | 1555 |
| 7.90 | 988 | 1153 | 1318 | 1483 | 1647 | 929 | 1084 | 1239 | 1394 | 1548 |
| 8.00 | 985 | 1149 | 1313 | 1477 | 1641 | 925 | 1079 | 1234 | 1388 | 1542 |

release temperatures commonly obtained as results of the explosion pressures and temperatures stated at the heads of the columns and the expansion ratios given in the side column of each table. These tables are based on values of 1.29 and 1.32 for the exponent $n$ of the expansion curve.

## Mean Effective Pressure

The mean effective pressure of a complete four-stroke cycle is the difference between the average pressure during expansion and the average pressure during compression, if we ignore the small loop of the diagram due to the formation of a partial vacuum behind the piston during the suction stroke. This loop is usually so small in comparison with the work area of the diagram that it is not worth considering.

**Table 6. Probable Mean Effective Pressures**

| Fuel | Engine H. P. | Compression Pressure, Pounds per Square Inch, Absolute | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | | 65 | 75 | 85 | 100 | 115 | 130 | 145 | 160 |
| Suction Anthracite Producer Gas | 10 | ..... | ..... | ..... | 55 | 60 | 65 | ..... | ..... |
| | 25 | ..... | ..... | ..... | 60 | 65 | 70 | 75 | ..... |
| | 50 | ..... | ..... | ..... | 65 | 70 | 75 | 80 | 80 |
| | 100 | ..... | ..... | ..... | 70 | 75 | 80 | 85 | 85 |
| | 250 | ..... | ..... | ..... | 75 | 80 | 85 | 90 | 90 |
| | 500 | ..... | ..... | ..... | 80 | 85 | 90 | 90 | 90 |
| Mond Producer Gas | 10 | ..... | ..... | ..... | ..... | 65 | 65 | 65 | ..... |
| | 25 | ..... | ..... | ..... | 60 | 65 | 65 | 70 | 75 |
| | 50 | ..... | ..... | ..... | 65 | 70 | 70 | 75 | 80 |
| | 100 | ..... | ..... | ..... | 65 | 70 | 75 | 80 | 85 |
| | 250 | ..... | ..... | ..... | 70 | 75 | 80 | 85 | 90 |
| | 500 | ..... | ..... | ..... | 75 | 80 | 85 | 90 | 90 |
| Natural and Illuminating Gases | 5 | 60 | 65 | 70 | 70 | ..... | ..... | ..... | ..... |
| | 10 | 60 | 65 | 70 | 75 | ..... | ..... | ..... | ..... |
| | 25 | 65 | 70 | 75 | 80 | 85 | ..... | ..... | ..... |
| | 50 | 70 | 75 | 80 | 90 | 90 | ..... | ..... | ..... |
| | 100 | 75 | 80 | 85 | 90 | 95 | 100 | ..... | ..... |
| | 250 | 80 | 85 | 90 | 95 | 100 | 105 | ..... | ..... |
| | 500 | ..... | ..... | 95 | 100 | 105 | 110 | ..... | ..... |
| Kerosene Spray | 5 | 50 | 55 | 60 | 65 | 70 | ..... | ..... | ..... |
| | 10 | 55 | 60 | 65 | 70 | 75 | ..... | ..... | ..... |
| | 25 | 60 | 65 | 70 | 75 | 80 | ..... | ..... | ..... |
| | 50 | 65 | 70 | 75 | 80 | 85 | ..... | ..... | ..... |
| Gasolene Vapor | 5 | 70 | 75 | 80 | 85 | ..... | ..... | ..... | ..... |
| | 10 | 75 | 80 | 85 | 90 | ..... | ..... | ..... | ..... |
| | 25 | 80 | 85 | 90 | 90 | ..... | ..... | ..... | ..... |
| | 50 | 85 | 90 | 95 | 95 | ..... | ..... | ..... | ..... |

In the two-stroke diagram there is no negative loop, but the mean effective pressure of the pump diagram should be subtracted from that of the work diagram to get at the true mean effective pressure of the cycle. As the pump delivery pressure is usually from 4 to 8 lbs. above the atmosphere, the mean effective pressure of the pump diagram is not negligible.

Since the mean effective pressure of the power diagram depends on the mean pressures of the two curves and both of these involve many uncertainties, it is impossible to calculate the mean effective pressure with as close an approach to accuracy as can be done for a steam engine. The most one can do is to assume average conditions and base the estimate of mean effective pressure, or of the horse power which it produces, on those conditions. Even then it is necessary to know just what quality of gas or oil will be used and what the proportion of air to gas or to oil vapor is going to be, in order to make the roughest estimate of the mean effective pressure or the horse power. Table 6 gives average mean effective pressures for engines of different sizes, working with compression pressures of 65 to 160 lbs. absolute, and different classes of fuel, the most effective mixture of fuel and air being assumed. These figures are merely a rough general guide; the actual mean effective pressure obtained with any combination of the stated conditions might be considerably higher or very much lower than the pressure value given in the table.

## III

## COOLING AND HEAT LOSS

As explained in the discussion of combustion pressures and temperatures, the maximum absolute temperature of the gases in a gas-engine cylinder may be as high as 3,000 degrees, and is ordinarily around 2,300 to 2,500 degrees; a thermometric temperature of 2,000 degrees is quite common. A moment's reflection will convince the reader that if the cylinder walls were allowed to reach any such temperature operation would be impossible. Lubricating oil would be instantly decomposed, and, even disregarding the effect of the heat upon the iron itself, the piston would soon jam by lack of lubrication.

In order to make it possible for such enormous temperatures to exist intermittently in the cylinder without preventing operation, the escape of heat through the cylinder walls is assisted by jacketing the cylinder and passing water through the jacket space continuously. Reference to Figs. 1, 2 and 4 will show that the cylinders illustrated are provided with double walls; the outer of these walls constitutes the water jacket. In Fig. 1 the water in the jacket is also represented.

The water is usually admitted to the jacket of a horizontal engine at the bottom and near the combustion end of the cylinder, and is discharged at the top near the outer end of the piston travel. The object of this arrangement is to apply the coldest water at the hottest part of the cylinder wall, which is that part surrounding the clearance space, where combustion occurs. Moreover, as the water becomes heated it tends to rise, and this tendency is assisted by putting the entrance at the bottom and the exit at the top of the jacket.

The circulating water, or jacket water, as it is commonly called, usually carries away from one third to one half of the heat liberated by the combustion of the gases in the cylinder. The ideal operation would be to proportion the water flow so that the amount of heat taken away will be just sufficient to keep the surface temperature of the cylinder walls and piston below the point at which the lubricating oil will burn or decompose. Of course, the temperature of the cylinder-wall surface cannot be measured directly, but an experienced engine runner can judge pretty well as to how the heat inside is affecting the cylinder oil by watching the exhaust and listening to the working of the piston.

The faster the water passes through the jacket, the more heat will it take away and the lower will be the temperature of the water at the point of discharge, if the initial temperature remains unchanged; or, at a given rate of flow, and a given quantity of heat taken out per minute or per hour, the discharge temperature will depend entirely on the temperature of the water as it enters the jacket. Consequently, no hard-and-fast rule can be laid down as to the temperature which the water should show as it emerges from the jacket. Ordinarily, this temperature is from 50 to 100 degrees above the temperature at which the water entered the jacket.

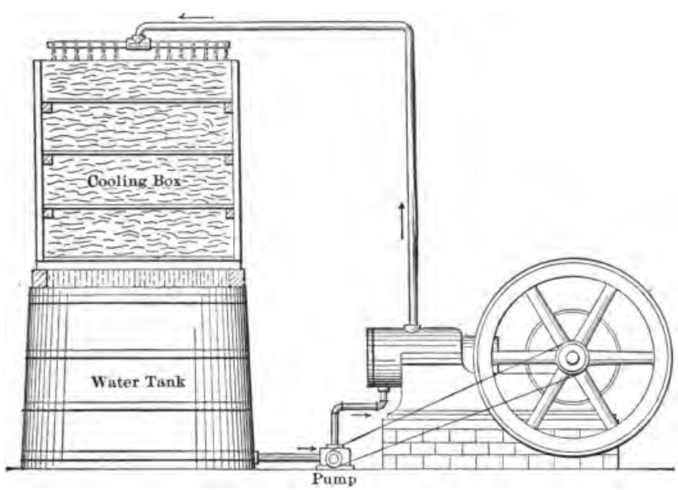



FIG. 11. — COOLING SYSTEM WITH PUMP.

As a rule, the cooling water is pumped through the jacket, then cooled off and returned to the pump by gravity to be forced through again. Such an arrangement is illustrated elementarily in Fig. 11. In the case of a small engine, however, it is often unnecessary to use a pump. The heating of the water produces a circulation sufficient to keep the temperature within bounds. An elementary arrangement of this sort is illustrated in Fig. 12. In both illustrations the arrows indicate the course of the cooling water. In Fig. 12 it will be noticed that the delivery pipe *a*, leading from the water jacket of the engine to the cooling tank, enters the tank below the water level. This is necessary in order that the height of water in the tank may be equal to the height of water in the vertical pipe and engine jacket, up to the horizontal pipe *a*. The water in the tank is cooler than that in the vertical pipe and engine jacket, and therefore heavier; consequently, it forces the hot water upward and out of the pipe *a*. The level in the tank does not need to be greatly higher than the outlet of the pipe *a*; it is wise, however, to keep it several inches above the outlet of the pipe in order that evaporation of water may not bring the level down far enough to stop the circulation before it is noticed. In order to prevent air locks and to allow any water vapor to escape without going into the tank, a vent *b* is provided. The cocks *c* and *c* are for the purpose of shutting off the tank whenever it may be necessary to disconnect the circulating pipes, and thereby avoiding the emptying of the tank. The valve *d* is to drain all the water from the engine jacket and the connecting pipes when the engine

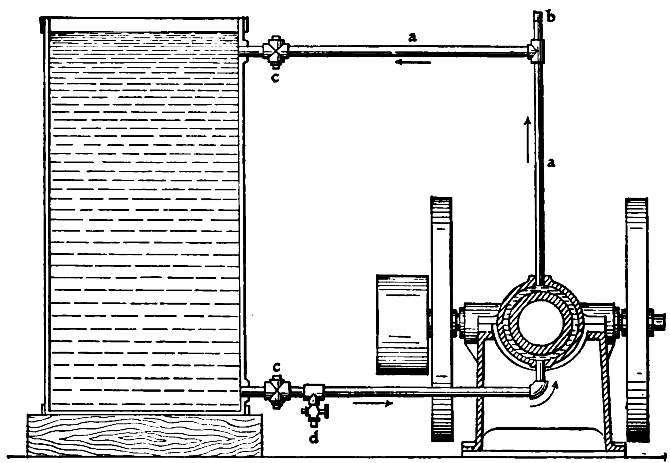



FIG. 12. — COOLING SYSTEM WITHOUT PUMP.

is to be shut down in very cold weather; then the cocks *c* and *c* are closed, of course.

In some small engines the excess heat is disposed of by means of thin flanges on the outside of the cylinder, which is made with a single wall, of course; the heat is radiated from these flanges to the surrounding air and carried away by the natural air currents or by a stronger draught produced by a fan blowing against the cylinder. Fig. 13 is a sectional view of a small vertical engine of this type. Air cooling has been found impractical with cylinders of more than 5 inches diameter of bore, and the usual limit is 4 inches.

Double-acting engines are always provided with hollow pistons of the short type and hollow piston rods and tail rods, and cooling water is forced through these as well as the cylinder jackets. This is necessary because both ends of the cylinder are closed and explosions occur in both ends; consequently, there is no opportunity to radiate any heat to the outer air from the piston, and the number of explosions per revolution of the crank shaft is twice as great as in the single-acting engine, giving

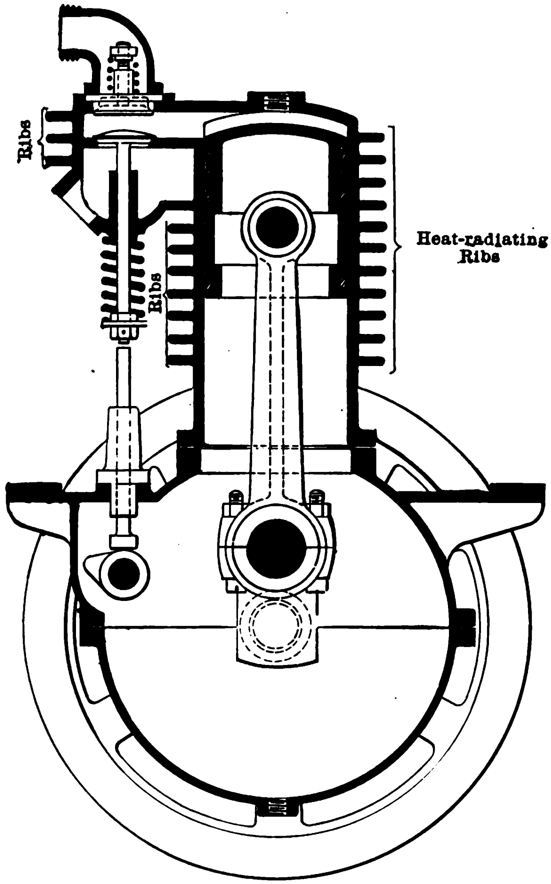



FIG. 13. — A VERTICAL AIR-COOLED ENGINE.

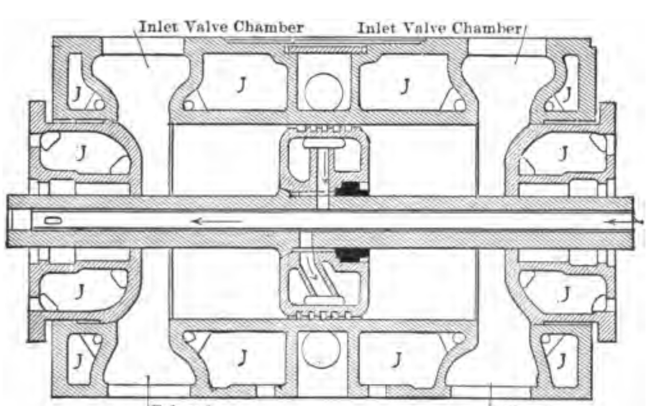



FIG. 14. — CYLINDER AND PISTON OF A DOUBLE-ACTING ENGINE.

twice the quantity of heat to be got out of the cylinder per revolution. Fig. 14 illustrates this construction of piston and rods, and the cylinder construction of a well-known German double-acting engine. The spaces marked $J$ are all connected together and constitute the water jacket of the cylinder. The inlet and exhaust valves are omitted in order to simplify the drawing.

## IV

## VALVES AND VALVE GEAR

THE inlet and exhaust valves used in gas and oil engines are of the poppet type for the reason that the high temperatures to which the valve faces and seats are subjected make the use of sliding valves impracticable. Since it is impossible to connect the stem of a poppet valve rigidly with the valve gear and adjust the connection so as to seat the valve accurately when it is closed, it is the universal practice to mount the valve in a "cage" which carries a guide for the valve stem, provide a spring which will constantly press the valve toward its seat, and arrange the valve gear so that a push rod or the end of a lever will press against the outer end of the valve stem at the proper moment and lift the valve from its seat. This sort of construction is illustrated in Fig. 15, where *E* indicates the exhaust valve and *I* the inlet valve. The exhaust valve is opened by the rocker arm *A* when the long end of the arm is pushed back by the valve rod, the latter being moved by a cam not shown in the drawing. When the cam

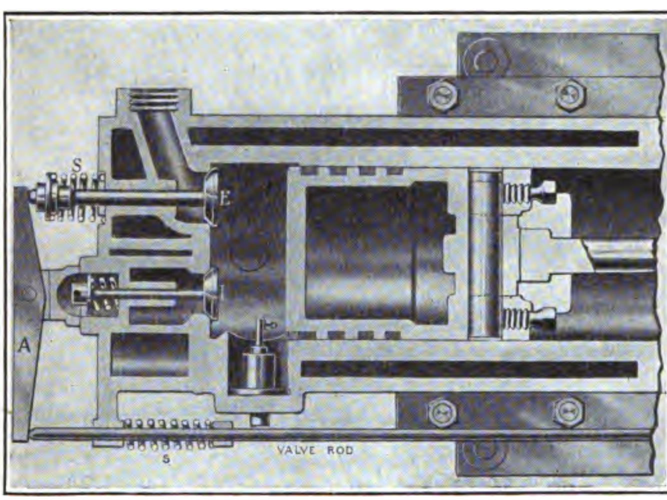



FIG. 15. — INLET AND EXHAUST VALVES; EXHAUST-VALVE ROCKER ARM AND PUSH ROD.

releases the unseen end of the valve rod, the spring *s* slides the rod back away from the rocker arm *A* and the heavier spring *S* presses the valve back to its seat. In this engine, the inlet valve *I* is of the "automatic" type; it is held on its seat by a weak spring and when the piston moves forward on the suction stroke, forming a partial vacuum behind it, the pressure of the atmosphere opens the valve.

The valve rod, in the case of the exhaust valve illustrated in Fig. 15, is pushed to the left by a rocker arm and cam arranged as shown in Fig. 16. The shaft on which the cam is mounted is driven by gears from the crank shaft of the engine, which works on the four-stroke cycle; as the

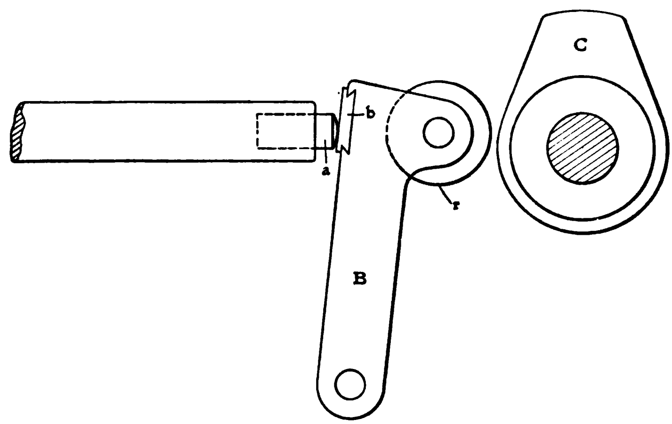



FIG. 16.—CAM, ROCKER, AND PUSH ROD.

exhaust valve is to be opened only once every two revolutions of the crank shaft, the cam shaft is geared down to one half the speed of the crank shaft. (This is true of the valve-gear shaft of every four-stroke-cycle engine.) The ends of the valve rod in Fig. 15 are provided with hardened steel buffer pins, as indicated at *a*, Fig. 16, and the rocker arms are fitted with corresponding hardened steel plates, as at *b*. The rocker arm *B* is also provided with a hardened roller *r* against which the cam *C* presses in opening the valve; the object, of course, is to reduce the friction as much as possible. This is only one of many forms of valve-operating mechanism, but nearly all are based on the general principles embodied in the form shown.

Some large engines are equipped with eccentrics instead of cams on the half-speed shaft, but some form of cam must be interposed between the eccentric rod and the valve stem, as a rule. Fig. 17 illustrates an American valve gear of this class in which one eccentric operates both the inlet and the exhaust valve. The motion of each eccentric rod is transmitted to its valve stem by a pair of "wiper cams." In the drawing, the valve gear is shown in that position where the inlet wiper cams are farthest apart and the exhaust wipers are at the "full open" position. This mechanism has the advantage of opening the valves and allowing them to close without serious shock or pounding. When the wiper cams first

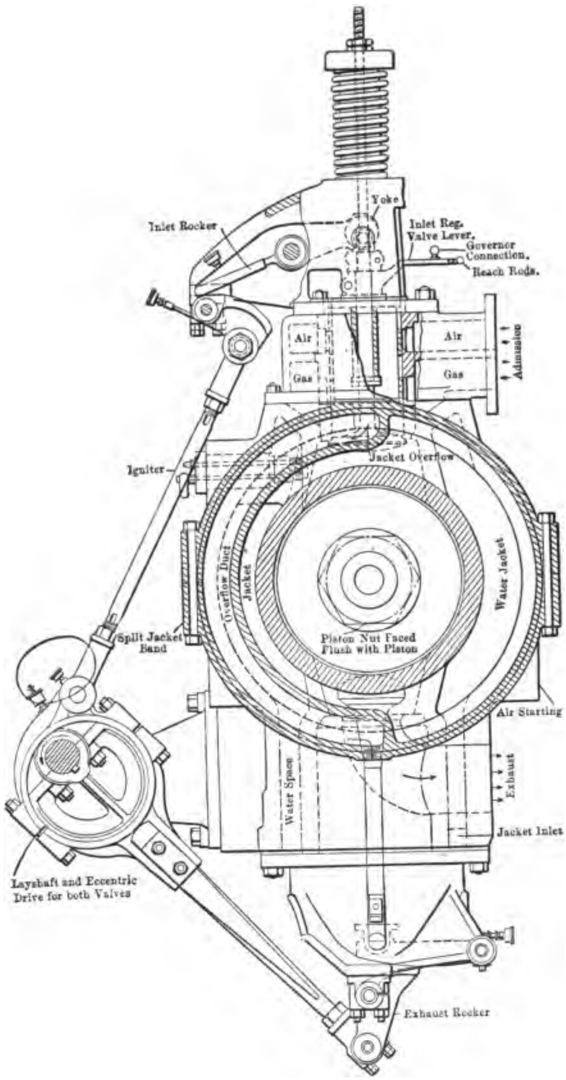



FIG. 17.—ECCENTRIC AND WIPER-CAM VALVE GEAR.

engage, the leverage is all in favor of the eccentric-rod cam, which is moving much faster at its tip than the valve-stem cam first moves at its tip; consequently, the valve is started from its seat very slowly and the pressure against the valve mechanism beyond the wipers is much less than if a direct cam movement were employed as in Fig. 16. After the valve is off its seat, and the pressure on its disk relieved, its motion becomes quicker with relation to that of the eccentric rod and it reaches the "full open" position very quickly. In closing, the reverse relations are true; the leverage is in favor of the valve stem, and the valve begins to close rapidly. As it nears its seat, the leverage having shifted back again, its motion is retarded so that when it is finally released by the cams it is seated with the force due to the elasticity of its spring only, instead of having high momentum to increase the seating force.

### The Mixing Valve

Besides the exhaust valve and the main inlet valve, there is frequently employed what is termed a "mixing valve," the function of which is to produce a more intimate mingling of the particles of gas and air than would occur if the two constituents were merely turned into the inlet

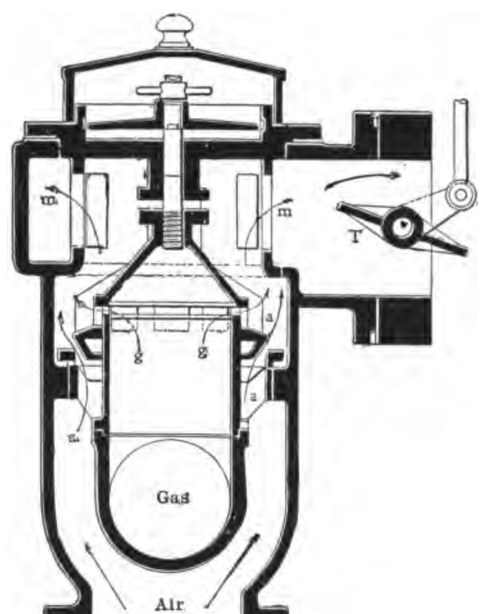



FIG. 18. — ONE FORM OF MIXING VALVE.

passage together from their respective sources. The general principle of all mixing valves is the same; the object is to break up the stream of gas into particles and to mix those with the particles of the incoming stream of air, and the general method is to force the gas and air to pass through a number of small openings together and to change their direction of flow abruptly in passing through these openings. Fig. 18 illustrates the form of mixing valve in use by one of the prominent American builders. The air-supply pipe is enlarged near the engine intake and a cylindrical chamber is secured in the center of this enlarged part; the gas is led in to the central chamber and passes from it into the air pipe through slots, as indicated by the arrows *g*, *g*; in the air pipe it meets the stream of air coming up from below as indicated by the arrows *a* and *a*, and the air and gas together flow into the engine intake passage through narrow ports in the upper end of the vertical pipe, as indicated by the arrows *m* and *m*. The butterfly valve *T* is a throttle, controlled by the governor to regulate the quantity of mixture admitted to the cylinder.

In all gas engines, provision is made for adjusting the proportion of air to gas in the mixture. Almost always the adjustment is made by means of a hand valve in the gas-supply pipe, the flow of air being unimpeded until the mixture reaches the throttle, if there be a throttle. On some engines, however, a special valve is used which simultaneously throttles the gas supply and opens up the air-supply passage to make the mixture "poorer," and *vice versa*.

# V

## IGNITION

In early gas engines the mixture was ignited by means of a "hot tube," consisting of a tube of relatively small diameter opening into the cylinder at one end and closed at the other; this tube being located within a housing and kept hot by a Bunsen burner. The compression stroke of the piston forced mixture up into the tube and the heat of the latter fired the mixture. During the suction stroke the burned gases in the tube expanded and prevented the fresh mixture from entering it. This arrangement has the serious disadvantages that any change in the relation between the length of the tube and the compression of the engine will change the timing of ignition.

It is almost universal practice now to ignite the mixture in the cylinder of a gas engine by means of an electric spark. Ignition in most gasolene engines is accomplished by this means also, but there are many kerosene engines in which other methods are employed.

### Make-and-Break System

There are two general systems of electric ignition: the "make-and-break" and the "jump-spark" systems. In the former, which is used in most stationary engines, electric current is passed through two separable contacts located in the cylinder and connected in series with an inductive (not induction) or "sparking" coil, and at the proper instant the contacts are separated by a mechanism operated from the valve-gear shaft. Upon the separation of the contacts, the current forms a spark between them, and this spark is greatly enhanced by the inductive coil in the circuit. Fig. 19 is a diagram of the ignition circuit of this system in which the separable contacts *A* and *B* are of the hammer type, held in contact with each other by a spring; *T* is a trigger attached to the same spindle as the pivoted contact *A* and located with its tip in the path of a trip rod *F*, which is lifted at the proper moment by a cam or some other form of mechanism mounted on or operated from the valve-gear shaft. As the rod *F* is lifted, the stop *J* presses it away from the end of the trigger *T*, allowing the latter to be drawn back into place by the helical spring shown. The end of the rod *F* is constantly pressed toward the stop *J* by the flat spring *S*. Just before the rod *F* is lifted to move the pivoted contact *A* away from the stationary contact *B*, the electric circuit is closed by the cam *C* wiping against the spring *D*, and the contacts are separated while the cam and

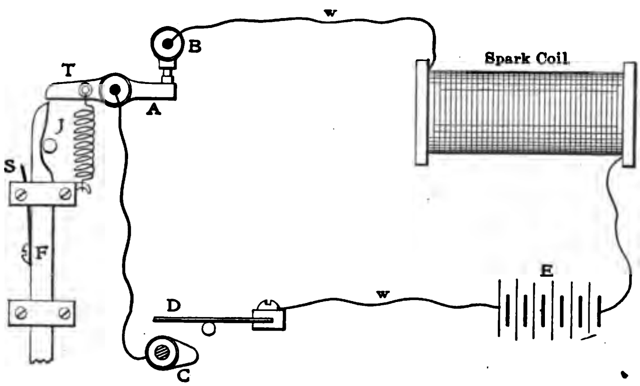



FIG. 19. — ELEMENTARY DIAGRAM OF A MAKE-AND-BREAK IGNITION SYSTEM.

spring are in contact and the circuit closed. When the contacts separate, opening the circuit, the spark coil gives an inductive "kick," and produces a much larger spark at the contacts than would occur if it were not in the circuit. The contact pieces *A* and *B* are, of course, located inside the cylinder; the stud of *B* and the spindle of *A* pass through an insulating block to the outside, where the electrical connections are made and the trigger *T* is attached. The source of current is indicated diagrammatically

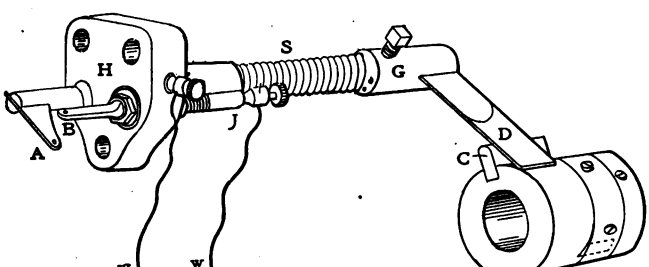



FIG. 20. — IGNITER OF A MAKE-AND-BREAK SYSTEM.

at *E* as an electric battery, but small electric generators are largely used for this purpose.

In most make-and-break systems the circuit-closing cam and spring *C* and *D* are unnecessary, the circuit being first closed and then opened by the sparking contacts. Fig. 20 illustrates the elementary principles of such a system, in which the rocking member *A* is brought into contact with the stationary terminal *B* by means of the cam *C* and tongue *D*, a coil spring *S* being interposed between the hub *G* and the spindle of the contact *A*. The use of this spring makes it possible to insure firm contact between the tips *A* and *B* without extremely accurate proportioning of the cam *C* and corresponding setting of the hub *G*; it also gives a sudden separation of the contacts when the cam passes the tip of the tongue *D*, and this enhances the action of the inductive or spark soil.

The wires *w* and *w* lead to the spark coil and the battery or generator, and correspond to the wires *w*, *w* in Fig. 19. The block *H* is made of metal, and the spindle of the contact *A* is not insulated from it; consequently, the current can pass from the binding post through the block *H* to the contact

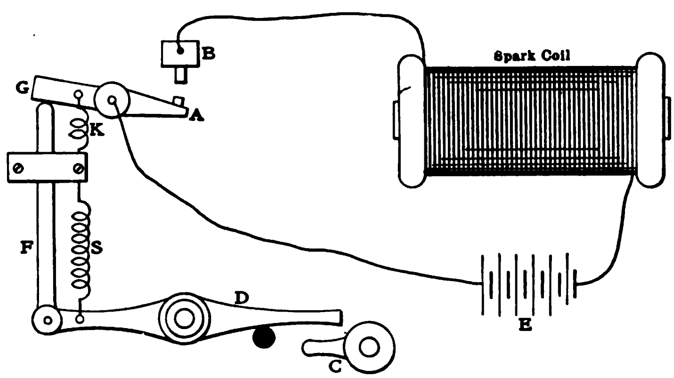


FIG. 21.

piece *A*. The stud *J* of the electrode *B* passes through a bushing of insulating material such as vitrified porcelain, so that there is no direct electrical connection between *A* and *B* when their tips are separated.

Fig. 21 illustrates the working principle of a class of make-and-break systems which is now more generally used than either of the two just described. In these systems two moving members are incorporated in the igniter proper, the rocking electrode and another lever which controls it, this actuating lever being operated by a cam, finger, push rod or other contrivance linked or geared to the valve-gear shaft, or else by an electromagnet. The rocking electrode *A* is normally held away from the stationary electrode *B* by the spring *S*, through the medium of the rocker *D* and push rod *F* acting on the lever *G*, which is fastened to the outside end of the electrode spindle. At the proper point in the cycle the finger *C*, mounted on a shaft which revolves in unison with the valve-gear shaft, carries the rocker *D* away from its stop, withdrawing the push rod *F* from the lever *G* and allowing the weak spring *K* to pull the rocking electrode into contact with the stationary one, closing the electric circuit and energizing the spark coil. An instant later the finger *C* rotates beyond the end of the rocker *D*, releasing it, and the spring *S* snaps the rocker back to the position shown; as it moves back, the push rod *F* strikes the lever *G* a sharp blow, knocking the electrodes apart very suddenly and producing the spark between their points.

The sketch does not indicate the construction actually used in any system known to the author; it only illustrates the principle of operation. In most cases the rocker *D* is mounted on a stud through the center of which the electrode spindle passes; a lateral lug on the actuating rocker takes the place of the push rod *F*. In some igniters a rotating cam is used, as at *C*; in others a reciprocating trigger is used, its end being pushed away from the rocker *D* by a roller partly in its path.

## Jump-Spark System

In the jump-spark system, an induction coil is used instead of a simple spark coil and the terminals inside the cylinder are both stationary, with their tips very close together so that the induced spark can jump across; the usual practice is to space the spark points or tips 0.03 to 0.05 of an inch

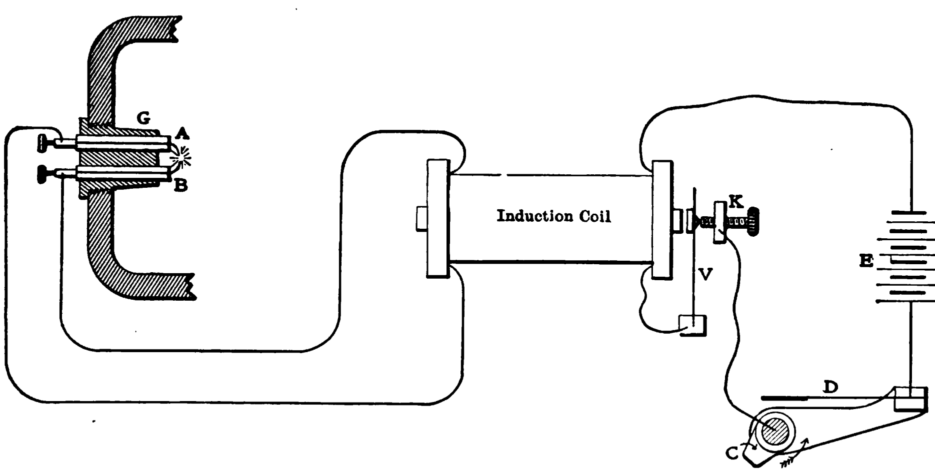



FIG. 22. — ELEMENTARY JUMP-SPARK SYSTEM.

apart, more often the smaller distance. Fig. 22 illustrates the principles of the jump-spark system. The sparking points *A* and *B* are mounted in an iron plug, *G*, from which they are insulated by porcelain bushings through which they pass. From these, wires extend to the secondary terminals of an induction coil. The primary winding of the induction coil receives current from a battery or other electrical source, *E*, through a circuit-closing cam and tongue *C* and *D* and a vibrator *V*. The latter consists of a steel spring carrying a soft-iron piece which is in line with the end of the iron core of the induction coil. When the primary circuit is open, the vibrator spring rests against a contact screw *K*, closing the circuit at that point. When the cam *C* comes into contact with the tongue *D*, the primary circuit is completed; the iron core of the coil is magnetized by the passage of current in the primary winding and attracts the vibrator *V*, drawing it away from the contact *K*, and opening the circuit there; then, the iron core being demagnetized by the cessation of current in the primary winding, the spring rebounds into contact with the screw *K*, closing the circuit again. These actions occur several times within the brief period of time while the cam *C* is rubbing against the tongue *D*, and the successive magnetization and demagnetization of the iron core of the coil induces a rapidly alternating electromotive force in the secondary windings; this pressure is so high that it readily jumps across the small air gap between the terminals *A* and *B* in the cylinder, and produces a series of flashes or sparks there which ignite the mixture.

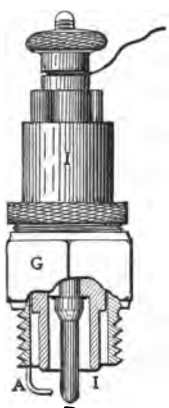


FIG. 22 *a*.—SPARK PLUG.

There are, of course, many variants of the two systems, but it is impracticable to describe here all of the arrangements actually in use. The jump-spark system as generally applied, however, does not include two insulated terminals or spark points in the cylinder. One of these is mounted in the metal plug and the other insulated from it, as illustrated in Fig. 22 *a*, where the point *A* is shown set into the plug *G* and the point *B* is insulated from it by the porcelain sleeve *I, I*. One of the wires from the secondary of the induction coil is attached to the outer end of the insulated sparking point, as shown in the drawing, and the other may be attached to any part of the engine that is in permanent metallic contact with the cylinder, because the plug *G* is screwed into the cylinder wall.

## Automatic Ignition

There are some oil engines which are not provided with any auxiliary apparatus for igniting the charge, because none is needed. The engine illustrated in Fig. 23 is one of these. It is designed to burn kerosene oil, which is injected by a small pump into a cylindrical chamber, known as a vaporizer, which is connected to the engine cylinder by a constricted passage, *N*. Air alone is drawn into the cylinder through the inlet valve during the suction stroke, and at the same time the oil pump discharges a jet of kerosene into the vaporizer, which is kept hot by the repeated explosions. The oil is vaporized by the heat of the chamber, and when the piston returns on the compression stroke, the air is forced back into the

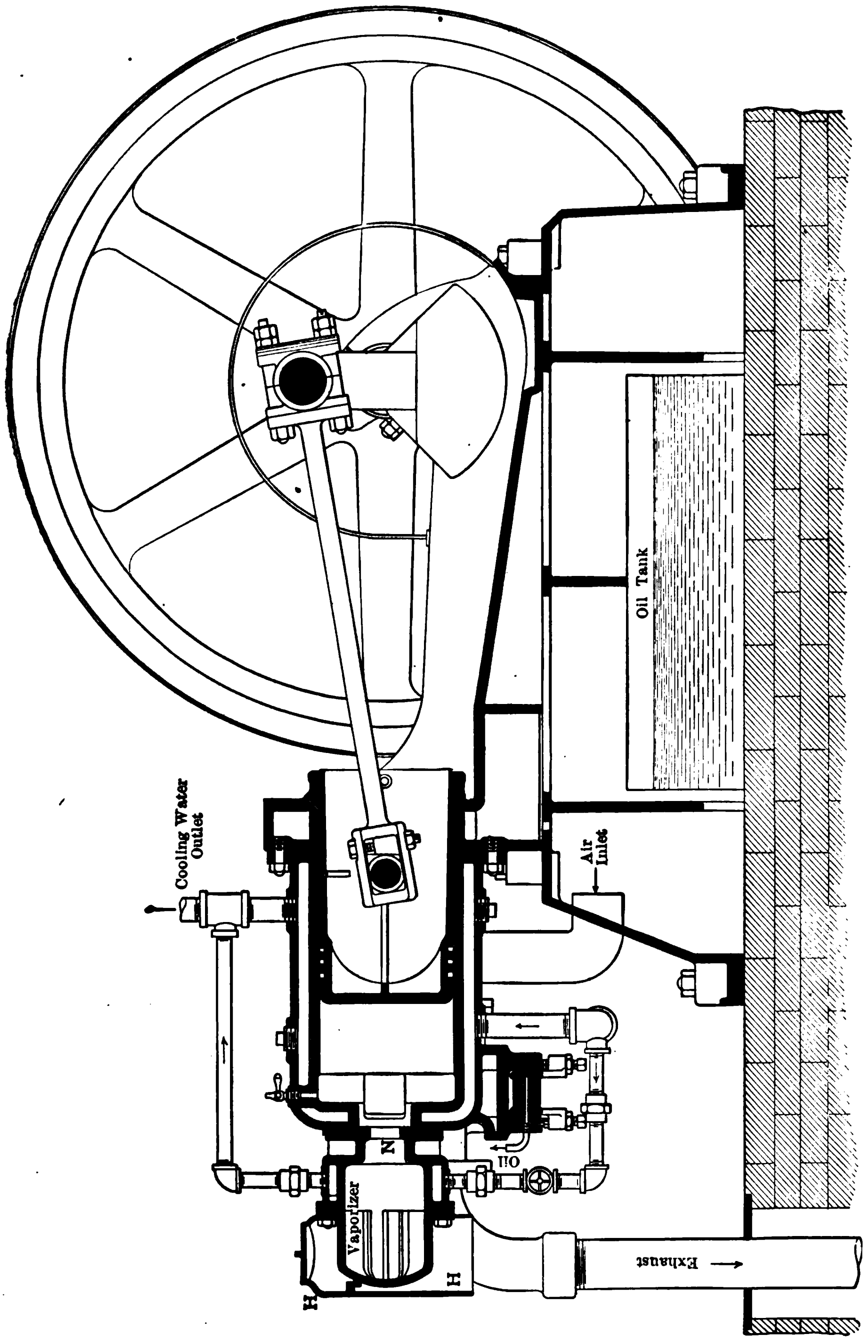



FIG. 23. — KEROSENE ENGINE WITH A VAPORIZER OPENING INTO THE CYLINDER.

vaporizer through the neck $N$, mixing with the oil vapor to form an explosive mixture. The compression of the air and the temperature of the vaporizer are so related that at the end of the compression stroke the mixture ignites by reason of the increase in temperature due to compression. The rear end of the vaporizer is covered by a hood, $H$, to prevent its being cooled by outside currents of air, while the end near the neck is water-jacketed like the cylinder. For starting the engine up cold, the vaporizer is heated by applying a blow torch or similar lamp at the mouth of the hood, beneath the vaporizer.

Fig. 24 shows another engine in which the oil is vaporized by the heat of a combustion chamber opening into the cylinder through a constricted passage. The neck of the firing chamber $V$ is extended on the lower side to form a lip $L$, on which the oil is sprayed through a nozzle $N$ at the beginning of the compression stroke. The lip is kept hot by the explosions and vaporizes the oil; the oil vapor and the air are forced back into the bulb $V$ by compression and the heat of the bulb added to that of compression fires the mixture.

In another kerosene-burning engine a thin metal plate is fastened by a small stud in the clearance space of the cylinder and a spray of oil is squirted on this plate through an atomizing nozzle by the oil pump, just as the piston completes the compression stroke. The plate, being thin and not in direct contact with any cooling medium, remains constantly at red heat, so that the oil is ignited as soon as it touches the plate, the clearance being filled with compressed air to support combustion.

Still another method of obtaining automatic ignition consists of compressing air alone in the engine cylinder to such a degree as to raise its temperature above the ignition temperature of oil, and injecting the oil into the clearance space in the form of a very fine spray. No hot plate is required because the compressed air is hot enough to ignite the particles of oil as fast as they are sprayed into the clearance space. The oil is forced in by means of a jet of compressed air supplied by a separate pump; delivery into the cylinder begins at the completion of the compression stroke and continues for a brief time after the piston has started on its power stroke.

## Timing the Ignition

Practically all gas engines and most gasolene engines are provided with mechanism whereby the moment of ignition can be adjusted through a considerable range, because it is usually necessary to ignite the mixture before the piston reaches the end of the compression stroke, and the extent to which ignition is advanced varies according to the character of the mixture, the speed of the piston, and other working conditions. It is impracticable to obtain a perfect mixture of the air and gas—that is, a mixture in which every atom of the gas is in contact with its proportion of the oxygen in the air, and there is no excess air; and a certain length of time, though very brief, is required for the flame to spread throughout

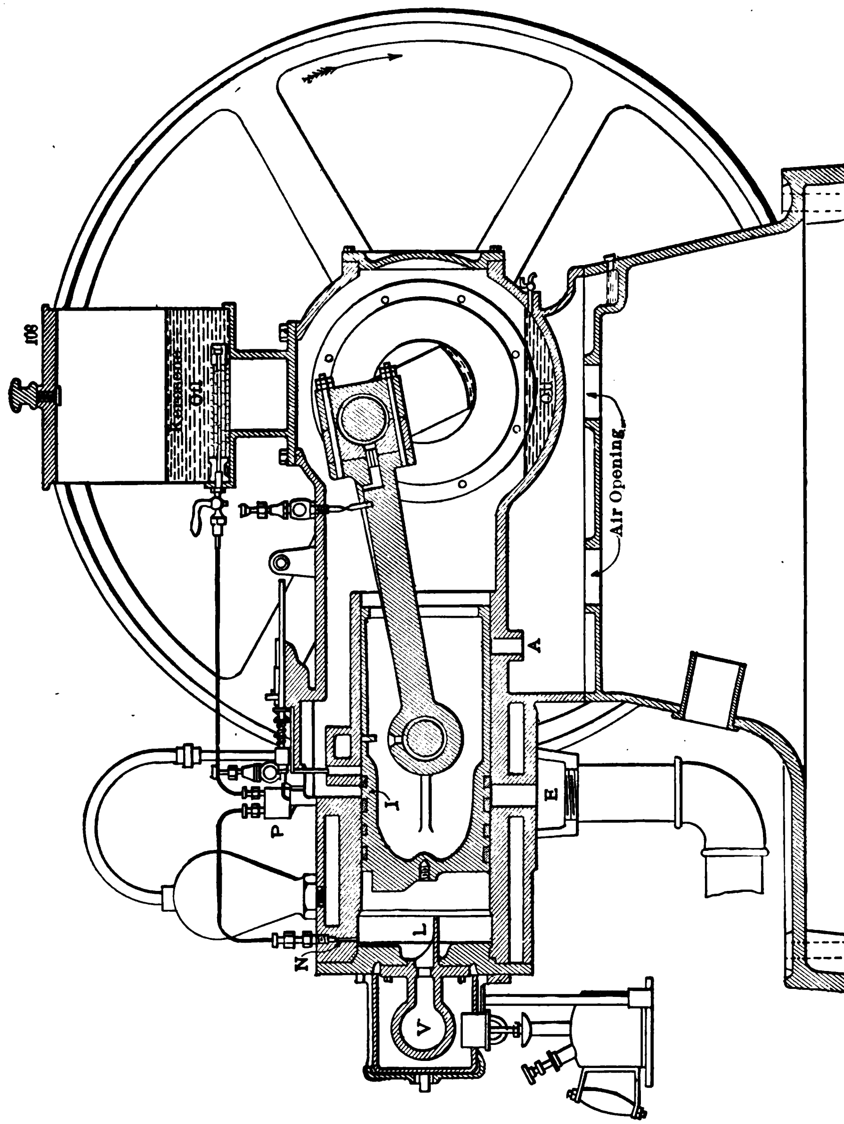



FIG. 24. — KEROSENE ENGINE IGNITING BY VAPORIZER-BULB HEAT.

the mixture, this time interval being increased by the imperfection of the mixture just mentioned. Therefore, in order that complete inflammation may be obtained at the desired moment (when the piston is at the end of the stroke) the mixture must be ignited a little before that moment.

The fraction of the piston stroke by which the moment of ignition is advanced depends chiefly upon the inflammability of the mixture and the speed of the piston. A thoroughly mixed and well-proportioned charge of gas and air will burn much more rapidly than a poorly mixed charge of correct proportions or a well-mixed charge of incorrect proportions, while a poorly mixed charge of unfavorable proportions will be comparatively sluggish in becoming fully aflame. Now, in order to get the most out of the fuel, the whole mass should be aflame when the crank is just over the exact dead center, and a mixture that is slow-burning must be ignited farther ahead of the dead-center position of the crank than one that burns more rapidly. For example, suppose that a certain mixture requires, under the conditions of operation, one two-hundredth of a second for the flame to spread throughout its entire mass after being ignited; then it should be ignited when the piston is at such a point in the compression stroke that one two-hundredth of a second will be occupied by the crank in reaching the position just barely over the exact dead center. In practice it is not necessary to measure the time required for a mixture to become completely aflame because the proper setting of the igniter can be readily ascertained by a little experimenting.

The "timing" of the igniting spark in gas and gasolene engines is highly important. If the mixture is ignited too soon, the rise of pressure due to combustion progresses too far before the crank reaches the dead center, producing wasteful and perhaps dangerous back pressure. If ignition occurs too late, the explosion pressure does not rise to the proper height and the power of the expansion stroke is reduced. Figs. 25 to 30 inclusive are reproductions of actual indicator diagrams taken from an engine running on natural gas and with the ignition varied from 25 per cent. ahead of the end of the compression stroke to 10 per cent. of the power stroke of the piston. That is, the diagram of Fig. 25 was taken with the igniter timed to produce the spark when the piston still had one fourth of its stroke to make before the crank reached the dead center, and Fig. 30 was taken with the igniter set to make the spark when the piston had traveled one tenth of the expansion stroke.

These diagrams illustrate very clearly the effects of different timing. In Fig. 25 ignition occurred so early that the combustion pressure attained its maximum before the end of the compression stroke, causing the negative loop at the top of the diagram. This is also true of Fig. 26, but to a less extent. In both cases pounding was caused by the premature rise of pressure, and this was so vicious in the first case that it would have been dangerous to continue running for any length of time.

In Fig. 27 the timing is excellent so far as smoothness of operation is concerned, but it was still too early to get the greatest power out of the fuel. In Fig. 28 this latter result was obtained, the mean effective

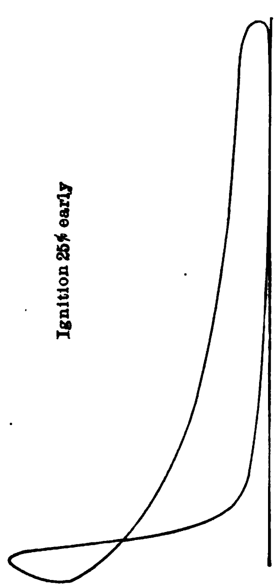



FIG. 25.

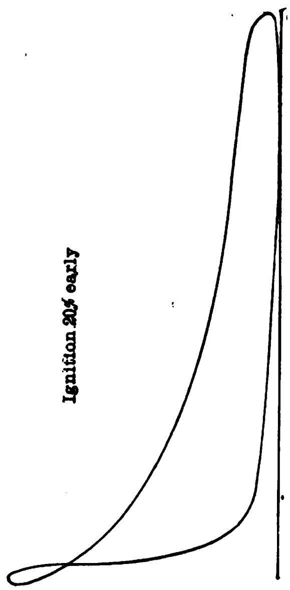



FIG. 26.

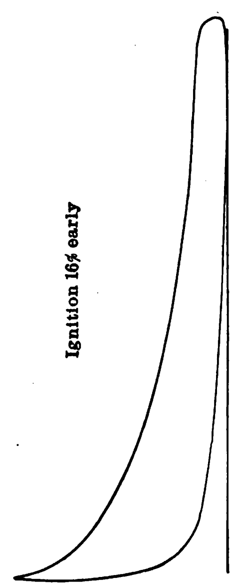



FIG. 27.

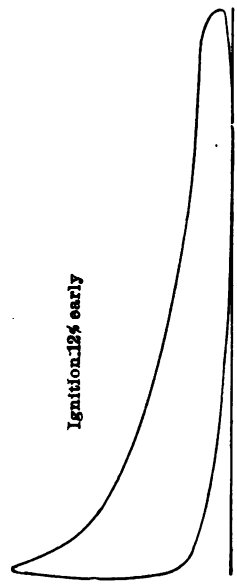



FIG. 28.

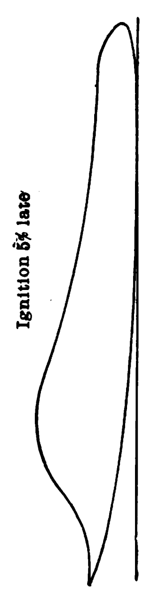



FIG. 29.

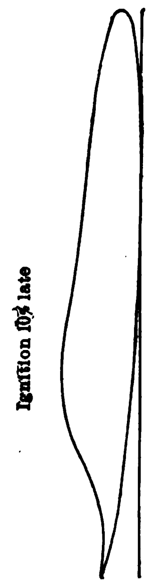



FIG. 30.

pressure being 93 lbs. per square inch. Curiously enough, this was exceeded by the diagram of Fig. 26, but the two diagrams were taken some time apart and it is more than likely that the higher mean pressure of Fig. 26 was due to a better combustible mixture.

Figs. 29 and 30 show the effects of late timing; these are a late rise of pressure and an unduly high exhaust pressure. In Fig. 30 the exhaust pressure was so high that the gases could not drop to atmospheric pressure until the piston had returned to about one fourth of its expulsion stroke.

The means of adjusting the time of ignition depends to a great extent upon the kind of ignition system employed. It is impracticable to describe, within the scope of this discussion, the different arrangements used in practice. The principle is the same in all; the position of one member of the igniter mechanism is made adjustable with respect to the member with which it coöperates in producing the spark which ignites the mixture. For example, in Fig. 19, the trigger *T* could be made adjustable on its spindle so that the upper end of the rod *F* would strike it sooner or later with respect to the dead-center position of the engine crank. In Fig. 20, the collar in which the cam *C* is set is adjustable around the shaft which drives it. In Fig. 21 the finger *C* would be made adjustable around its shaft or the gearing between this shaft and the valve-gear shaft would be adjustable to alter the angular relation between the two. In Fig. 22 the arm on the end of which the spring *D* is mounted is pivoted on the end of a journal box in which the cam-shaft revolves, so that the arm may be swung around, concentric with the shaft, and the moment at which the circuit-closing cam touches the spring thereby changed; this type of mechanism for timing the ignition is universally used with the jump-spark system, in both gas and gasolene engines.

The ignition point in most kerosene-burning engines is fixed by the builders and cannot be adjusted while the engine is running, because varying the time of ignition is not usually necessary with kerosene engines; the explanation of this is that the character of the mixture does not vary widely and, because of the high temperature required to vaporize kerosene, vaporization and ignition are commonly effected by contact with large hot surfaces or with the heated body of air in the clearance space, while in gas and gasolene engines the mixture is ignited at one point only and time is required for the flame to spread throughout the mass.

# VI

## MIXING LIQUID FUEL WITH AIR

THE principle of mixing gas and air by passing them through small openings and causing abrupt changes in the direction of flow, described in the chapter on valves, cannot well be applied to liquid fuels because the relative volume of the fuel is too small. In order to obtain anything like a thorough mixture, the oil must be either atomized—broken up into a fine mist—by means of a special nozzle or vaporized by means of heat. Gasolene evaporates at low temperatures, and it is therefore customary to vaporize it and mix the vapor with the air just before the charge enters

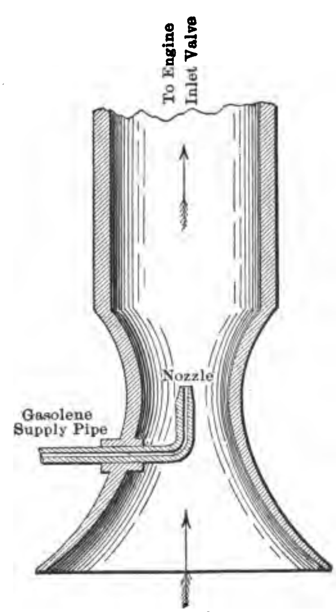



FIG. 31. — ELEMENTARY GASOLENE-AIR MIXER.

the cylinder. Kerosene requires to be raised to a rather high temperature for evaporation, and it is therefore the common practice to inject the liquid either into the cylinder or into a hot chamber connecting with the cylinder, as described in the preceding chapter, thereby securing vaporization without the application of a special heater, which would be required for mixing the charge entirely outside the cylinder.

Fig. 31 illustrates the general principle on which is based the operation of all efficient devices for mixing gasolene and air. The gasolene supply pipe terminates in a spraying nozzle located in the center of the air-intake pipe, not far from the inlet valve of the engine. A small pump

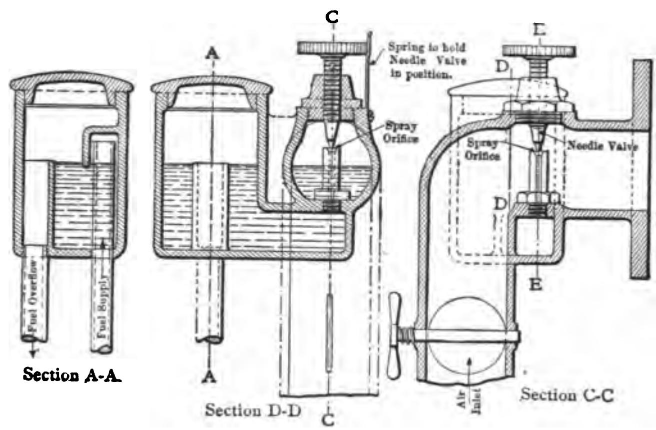



FIG. 32. — CONSTANT-LEVEL (OVERFLOW) MIXING VAPORIZER FOR GASOLENE.

driven from the valve-gear shaft forces a charge of gasolene out of the nozzle during the suction stroke of the engine, and the air that is being drawn into the cylinder catches up the gasolene spray, evaporates it, and carries the vapor along into the cylinder. The bore of the air pipe is usually reduced where the gasolene nozzle is located, as here shown, in order to increase the velocity of the air at that point and insure the picking up of all of the gasolene spray, but this construction is not always employed. The gasolene, being sprayed into the air current, is very thoroughly distributed, and the air is usually drawn from a point near the exhaust pipe of the engine, so that it is warm enough to vaporize the gasolene; the mixture, therefore, is very good. In a few forms of mixing vaporizer, however, a disk fan is pivoted in the air pipe just beyond the gasolene nozzle, and revolved by the rush of air past its blades; this gives the air and gasolene vapor a whirling motion and greatly increases the intimacy and efficiency of the mixture. This statement is based on actual tests made by the author with a well-designed vaporizer, applied with and without the fan.

Many gasolene engines are provided with a mixing vaporizer in which the gasolene is delivered to the spray nozzle at constant pressure, so that the fuel mixture may be regulated by fine gradations and with great accuracy. The constant pressure is obtained by supplying gasolene to the nozzle from a small chamber in which the gasolene is maintained at a constant level. This level is slightly below the level of the spray orifice, and the gasolene is drawn from the nozzle by the suction of the engine piston. Fig. 32 shows sectional views of such a vaporizer; the supply chamber is provided with an overflow pipe leading back to the main gasolene tank, and the upper end of this pipe is set at the level desired. The gasolene is pumped to the level chamber at a rate considerably higher than the engine requires, and the surplus goes back through the overflow pipe to the tank by gravity. A needle valve is provided at the spray orifice by which to regulate the quantity of gasolene drawn out by each

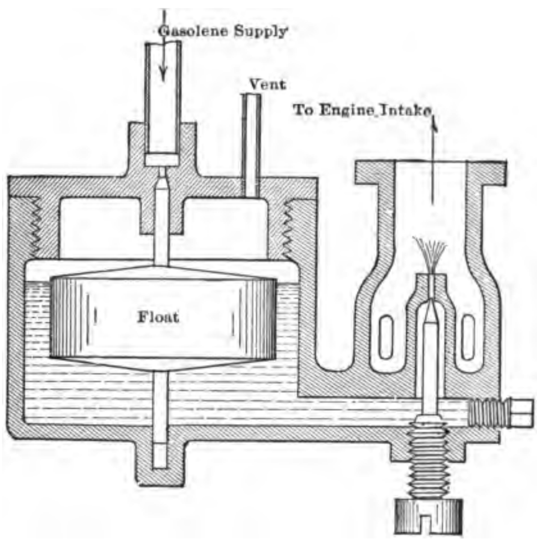



FIG. 33. — CONSTANT-LEVEL (FLOAT-VALVE) MIXING VAPORIZER FOR GASOLENE.

suction stroke of the piston. This valve also serves to spread the gasolene into a thin conical sheet and thereby promote its mixture with the air as it rushes by. The butterfly valve $T$ is a throttle, by which the charge taken in by the engine is regulated; the greater the quantity of air admitted by this valve, the greater will be its velocity at the spray nozzle and, consequently, the greater will be the quantity of gasolene sucked out, and *vice versa.*

Vaporizers of the type just described are practical enough for stationary engines, but not for engines used on automobiles and boats, because of the constant changes in position. For these classes of service the "float feed" type of vaporizer and mixer is almost universally used. Fig. 33 illustrates the general principle on which this type of vaporizer is based. The only important difference from the type in Fig. 32 is that the gasolene is maintained at constant level by means of a float valve. Gasolene is delivered from a main supply tank into a small chamber in which is a float, mounted on or attached to the stem of a needle valve controlling the inlet orifice. The float maintains a substantially constant level of gasolene in the apparatus, and this level is a little below the tip of the spray nozzle in the air pipe. The engine suction draws the gasolene out of the nozzle and the mixing is effected as in the previous case. The proportion of gasolene to air is adjustable by means of a needle valve located within the spray nozzle in this case. The float chamber is provided with a small vent, or more accurately, an equalizing orifice, through which atmospheric pressure may act on the gasolene and force it out when the air, rushing past the nozzle, forms a partial vacuum near it. This vent also serves to prevent the development of undue pressure in the reservoir by the evaporation of gasolene in very warm surroundings. Air enters the nozzle chamber through a series of holes in the wall below the level of the nozzle.

Very few kerosene engines built in this country are equipped with devices for vaporizing the oil and mixing it with the air outside the cylinder

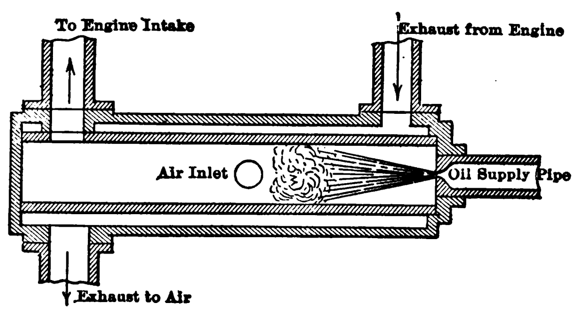



FIG. 34. — KEROSENE-AIR MIXING CHAMBER.

or some extension of the cylinder. One well-known kerosene engine, however, has a mixing vaporizer heated by the exhaust gases from the engine, of which Fig. 34 shows a longitudinal section. Air is admitted freely to the mixing chamber, and oil is pumped into it through an atomizing nozzle. The engine draws in the mixture by piston suction, as in the gas or gasolene engine. The mixing chamber is enveloped entirely by a jacket through which the exhaust gases from the engine pass to the exhaust pipe.

The next step is to vaporize the oil in a hot chamber opening permanently into the cylinder, as illustrated by Fig. 23, and described in the chapter on ignition. The third method, that of injecting the oil directly into the cylinder and vaporizing it by contact with a hot plate, was also described in that chapter.

Then there is a combination of the two methods just described; the engine in which it is applied is illustrated by Fig. 24. The oil is injected by a small pump *P* through the nozzle *N* into the cylinder and strikes the lip *L* of a hot bulb *V*. This occurs early in the compression stroke of the piston, and as compression progresses the oil vapor is forced back into the hot bulb *V*, where it and the air are compressed and then fired as in the engine illustrated in Fig. 23. The torch shown beneath the bulb *V* is for the purpose of heating the bulb before starting the engine. After it is running the explosions keep the bulb hot.

The fourth method is that employed in the Diesel engine, which works with very high compression. The piston draws air alone into the cylinder and compresses it to a pressure of several hundred pounds per square inch. The oil is blown into the cylinder through a suitable nozzle by a jet of compressed air, which breaks it up into a very fine spray, and the heat of the highly compressed air in the cylinder ignites the particles of atomized oil as rapidly as they enter. This combustion is not explosive, as in other internal-combustion engines, but gradual, continuing as long as the spray enters, and this period is varied by the governor according to the load requirements.

## VII

## METHODS OF GOVERNING

### Hit-and-Miss

There are three fundamental methods of governing the speed of a gas or oil engine and several combinations of these. The oldest is the "hit-and-miss" method, which consists of causing the engine to stop taking in charges of mixture when the speed drops back to normal. One way of accomplishing this on a gas engine is to provide a valve in the gas-supply pipe which is opened regularly by the valve gear during the suction strokes as long as the speed remains normal and allowed to remain closed when the speed exceeds the normal, the governor being arranged to control the actuation of the valve. The corresponding method with an oil engine is to arrange the governor so as to stop the pump or allow it to be operated, according to the speed. This application of hit-and-miss regulation to a gas engine is illustrated by Fig. 35. The rocker arm $A$ is moved by the cam $C$ at the proper moment to open the gas valve. To the left-hand end of the rocker arm is pivoted a finger, commonly called a "pick blade," and a link from the governor mechanism holds this pick blade either into or out of line with a hardened extension of the gas-valve stem. So long as the speed is normal, the governor keeps the pick blade in line with the valve stem, but if the speed exceeds the normal rate, the blade is drawn to the right far enough to miss the end of the valve stem when the rocker arm is operated by the cam; consequently, the gas valve remains closed during that suction stroke and the engine takes in air alone. There is no explosion, therefore, after the compression stroke is completed, and the engine speed is reduced by this failure to get a power impulse for the expansion stroke.

The principle embodied in the construction shown in Fig. 35 is applicable to an oil engine by merely substituting the oil pump for the gas valve, arranging the pick blade to strike the end of the pump plunger stem normally, and providing a spring to return the plunger after each delivery stroke. This arrangement, however, cannot be used with any form of vaporizer which contains a fuel reservoir, as in Fig. 32 or Fig. 33, because the missing of a pump stroke would not cause the engine to miss its fuel charge immediately. It is only practicable where a vaporizer is used without any constant level supply, as in Figs. 31 and 34, or where the fuel is injected directly into the cylinder or an extension of it, as in Figs. 23 and 24.

The application of the hit-and-miss method illustrated in Fig. 35 has the advantage that whenever the governor cuts out a charge the piston draws in pure air which mingles with the burned gases in the clearance space and greatly reduces the quantity of useless material remaining in the

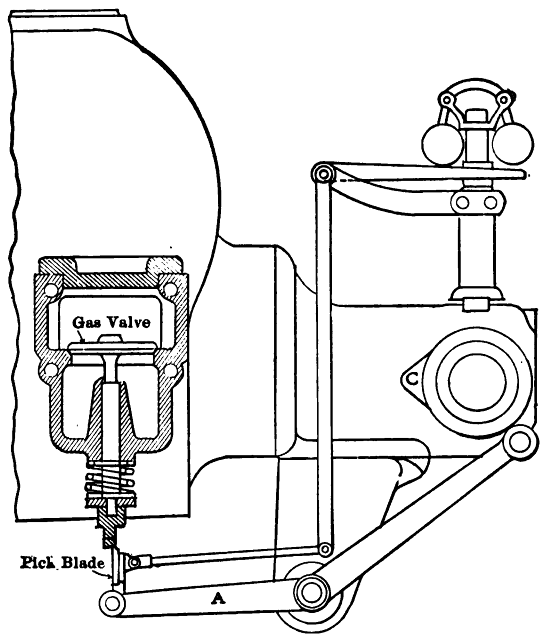



FIG. 35. — A HIT-AND-MISS GOVERNOR MECHANISM.

cylinder when the next charge is taken in. It also cools the cylinder contents and permits a greater weight of mixture to be taken in during the next effective suction stroke.

Under some conditions the features just mentioned may be disadvantageous instead of desirable. For example, if the engine is running with a rather small load, so that it misses more explosions than it gets, the cylinder may be cooled down sufficiently to impair the efficiency of the engine or even to cause it to discontinue firing the occasional charges, in the case of a kerosene engine; furthermore, if the mixture is adjusted for maximum effectiveness at full load when there are few or no "misses" and the clearance space is filled with hot burned gases almost every time the cylinder takes in a fresh charge, then the cylinder contents will not have maximum effectiveness at light loads when the clearance is filled with almost pure air at each admission of mixture and the temperature is lower. However, it is a simple matter to adjust the proportion of gas or oil to air when the engine is running at its average load; then the discrepancy at greater or smaller loads will not be so objectionable.

An application of hit-and-miss regulation that has been used to a considerable extent is illustrated elementarily in Fig. 36. The engine is equipped with an automatic inlet valve and the governor is arranged so that when the speed is too high it moves a detent *d* in position to catch a dog *e* on the exhaust-valve stem after the valve has been opened by the push rod, and thereby hold the valve open. The consequence is that when the piston moves forward on its suction stroke it cannot form a vacuum in the cylinder and the inlet valve is not opened. The cylinder, therefore, does not get a charge of fresh mixture, and misses the explosion that would

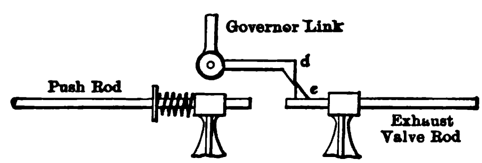



FIG. 36. — ANOTHER HIT-AND-MISS METHOD.

ordinarily next occur. This arrangement is open to the objection that when the governor blocks the exhaust valve open during a suction stroke the piston sucks in hot burned gases from the exhaust pipe. It has the advantage, however, of being applicable to oil engines having constant-level mixing vaporizers because, as the main inlet remains closed during the suction stroke, no fuel can be drawn in.

The hit-and-miss method of regulation has the sole advantage of high fuel economy. The engine takes in practically a full piston displacement of mixture every time it takes a charge, and the mixture is of uniform quality; consequently, it burns with high efficiency. On the other hand, the power impulses do not occur at regular intervals unless the engine is carrying the maximum possible load, and this necessitates a very heavy flywheel in order to prevent too great a drop in speed between impulses. Moreover, when running at moderate load the working parts are subjected to just as violent stresses due to the sudden rise of pressure after ignition as when the engine is carrying its maximum load.

## Variable Quantity of Intake

The next simplest method of speed regulation is that of throttling the mixture, just as a throttling steam engine controls the quantity of steam admitted to the cylinder. This is usually accomplished by putting a butterfly valve in the inlet pipe and controlling its position by the governor, as indicated in Fig. 18. The arrangement is so simple and obvious that further illustration of it is unnecessary. The throttle valve is usually located just as near the main inlet valve as possible, in order to obtain prompt response to a change in its position. With gas engines this is advisable for the additional reason that the quality of the mixture is maintained more nearly constant than it would be if the throttle were in the air-intake pipe.

The advantages of throttling the mixture are simplicity of mechanical construction and the maintenance of a practically constant ratio of gas or oil to air. The disadvantage is that the compression in the cylinder changes with the load. Thus, at light load, the governor admits a smaller quantity of mixture during each suction stroke than at full load, and the compression pressure is therefore lower. This means that the particles of fuel and air are not so intimately compressed, and the result is slower combustion. At very light loads it frequently occurs that the compression is so low that a charge is either not completely burned or not even ignited. In either case there is a loss of fuel, and when ignition fails to occur there is also likely to be an explosion in the exhaust pipe of the charge which passed unburned through the cylinder. This is called "after-firing" and sometimes produces disaster by wrecking the flue through which the exhaust gases pass to the atmosphere. The low compression at light loads also causes a serious falling-off in efficiency; this is discussed in the chapter dealing with the calculation of pressures, output, etc.

Another method of varying the quantity of mixture taken by the engine during each suction stroke is by closing the admission valve sooner or later in the suction stroke, according to the demands of the load—in other words, "cutting off" exactly as in the automatic cut-off steam engine. Fig. 37 illustrates a form of cut-off gear that has been applied to an engine having separate gas and air inlet valves. The two valves are on one stem, the air valve being rigidly mounted and the gas valve spring-mounted. The valve stem is depressed by the rocker arm *A* when its outer end is raised by the push rod, and the nose of the latter is pressed out from under the rocker arm by the tripping plate *P*, allowing the valves to be seated by the upper spring on the valve stem. The trip *P* is mounted on one arm of a bell crank which is tilted forward more or less by the governor, thereby determining the point of cut-off and consequently the amount of mixture taken in during the suction stroke. The push rod is moved up and down by an eccentric on the valve-gear shaft.

Fig. 38 shows another form of cut-off gear embodying the same principle as illustrated by Fig. 37. The air valve *A* and the gas valve *G* are connected rigidly, however, by a barrel *D*, which encircles the lower end of the spring that seats the valves. When the air valve is down on its seat, the gas valve closes the gas port, which is tapered, with a sufficient approach to absolute tightness to prevent appreciable gas leakage. The main inlet valve *I* is opened by a cam on the valve-gear shaft at the beginning of the inhalation stroke and held open throughout the stroke. The stem of the

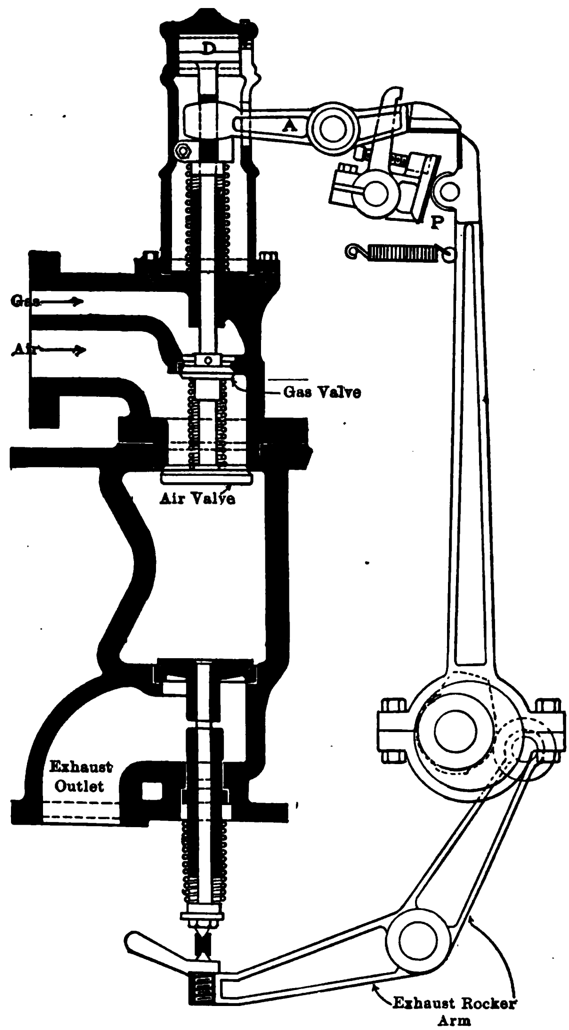



FIG. 37. — AN AUTOMATIC CUT-OFF GOVERNOR MECHANISM.

inlet valve is linked to a short rocker arm *R*, Fig. 38, to the other end of which is pivoted a block arranged to slide vertically in a guide. To this block is hung the pivoted latch *L* (shown in Fig. 39), the end of which normally engages a dog on the block which is screwed on the upper end of the stem of the cut-off valve. When the main inlet valve is opened, the latch

*L* lifts the cut-off valves, allowing air and gas to pass to the mixing chamber *M* through the ports shown closed by the valve disks *A* and *G*, respectively. From the mixing chamber the mixture passes into the cylinder through the port of the main inlet valve *I*; at the point in the suction stroke determined by the governor, the cam *C*, in Fig. 39, engages a lug and draws the drag link over, thereby pulling out the latch *L* and allowing

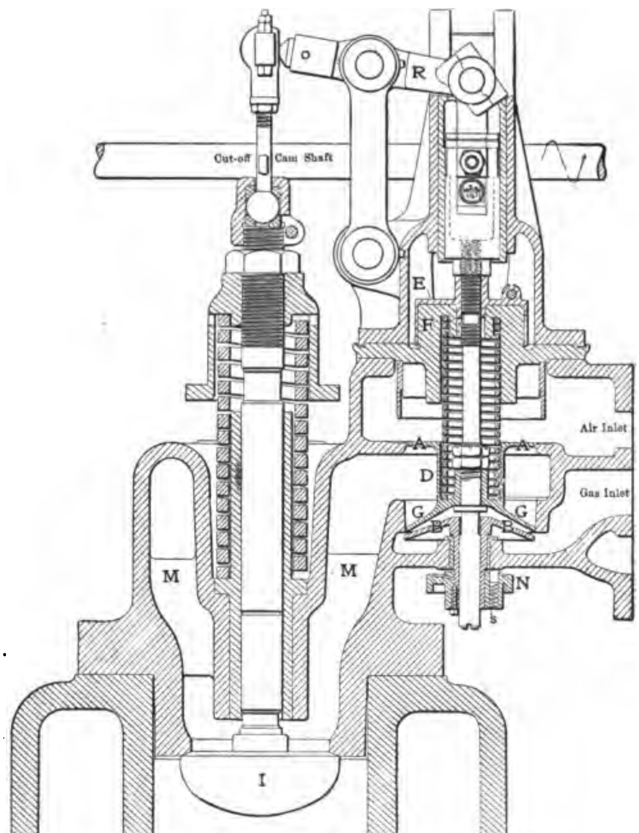



FIG. 38.—SECTIONAL ELEVATION OF INLET AND CUT-OFF VALVES AND MECHANISM.

the cut-off valve to drop. The drag link is pivotally attached to a lug on the latch *L*, and its other end is curved around the cut-off shaft *S*, the upper leg of the bend resting on the journal box and holding the link in place as it slides back and forth. Before the succeeding suction stroke begins, the cam *C* has turned to the "low" side and the latch *L* is thrown into engagement with the valve-stem dog by a small helical spring. When the cut-off valve drops, it is cushioned by the inverted cup *E* (Fig. 38), acting as a dash pot, the boss $F$ constituting the plunger. The cut-off cam shaft $S$ rotates continuously at the same speed as the main valve-gear shaft, and its angular position with respect to that of the valve-gear shaft is adjusted by the governor through a "floating" bevel gear.

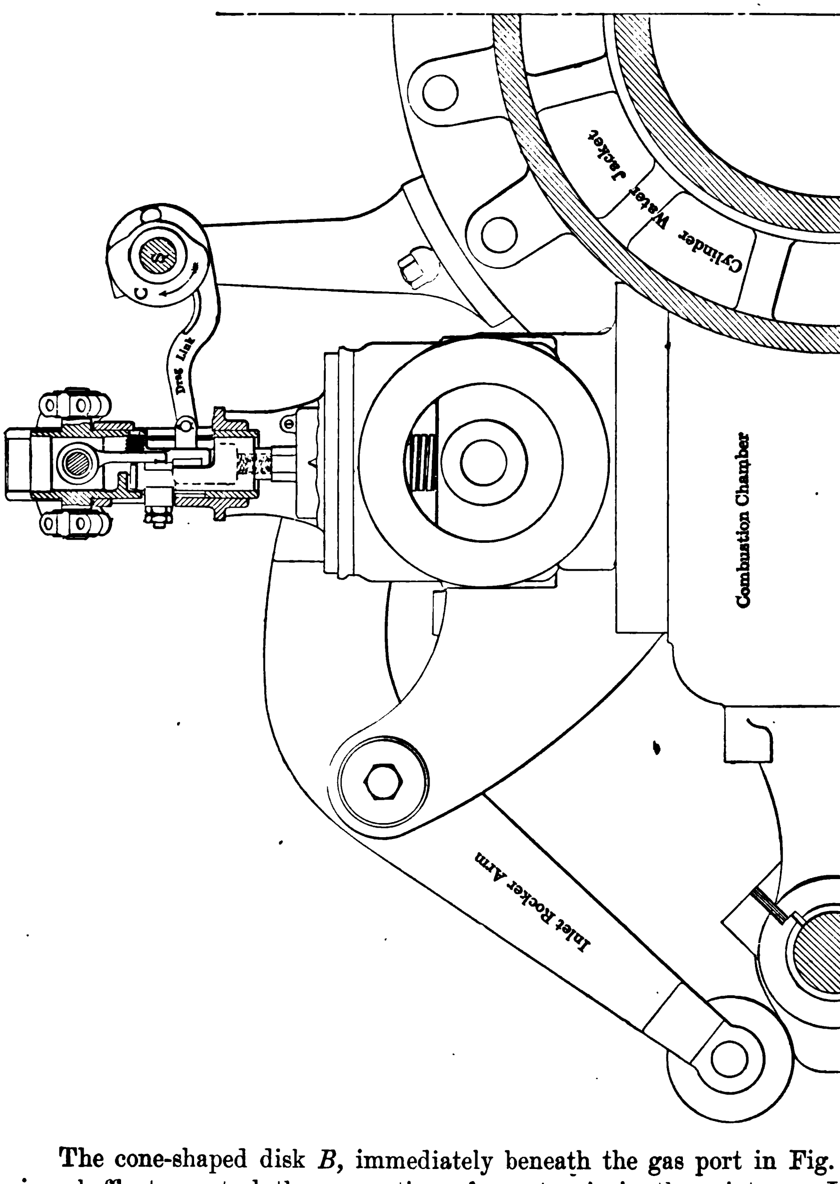



FIG. 39.—END ELEVATION OF INLET AND CUT-OFF VALVE MECHANISM.

The cone-shaped disk $B$, immediately beneath the gas port in Fig. 38, is a baffle to control the proportion of gas to air in the mixture. It is adjustable toward and away from the gas port by means of the knurled head $N$, which is locked in the proper position by the set screw $s$. The shank of the baffling disk serves also as a guide for the lower end of the stem of the cut-off valve disks $A$ and $G$.

The automatic cut-off method of regulating the speed involves the same advantages and disadvantages that were mentioned as features of the throttling method. There is a slight advantage, however, in cutting off as compared with throttling in that the negative work done by the piston during the suction stroke is a trifle less. This is due to the fact that the throttling engine at underloads draws in its charge past a partly closed throttle during the whole suction stroke, whereas the cut-off engine draws in its charge through wide-open passages and during less than the full piston stroke.

## Varying the Quality of Mixture

The third fundamental method of governing is to vary the quantity of fuel alone, keeping the total quantity of cylinder contents constant. This is done in gas engines by putting a throttle in the gas-supply pipe and controlling it with the governor, or by arranging a cut-off valve in the gas passage, opening it at the beginning of the stroke when the air valve opens and closing it earlier or later in the stroke by means of the governor. Both of these methods are open to the serious objection that the quality of the mixture is varied as the load changes. An engine so regulated will not work under widely varying loads because neither a very rich mixture nor a very poor mixture will ignite. Where the load does not vary much within short periods of time this method of governing is fairly satisfactory, but it involves constant watchfulness to see that the proportion of air to gas does not become too large or too small. The only advantage of "quality regulation," as it is called, is that the compression remains constant at all loads, giving better efficiency below full loads than if the compression varied.

This method is the one generally used on kerosene engines and on gasolene engines which feed the fuel to the cylinder with a pump. It is not used with constant-level vaporizers.

## Combination Methods

With a view to overcoming, avoiding or minimizing the disadvantages of the three fundamental methods, several combinations of them have been tried. One engine builder has used quality regulation at full load and for under-loads as far as possible, and at smaller loads than this method would handle the hit-and-miss method came automatically into play.

The most successful compromise thus far attempted, however, seems to be that of admitting air during the complete suction stroke and varying the time at which gas *begins* to enter, continuing the intake of gas with the air to the end of the stroke. Thus, if the load is such as to require the admission of gas during one half of the stroke, the gas valve remains closed until the piston has traveled halfway of the suction stroke and then it is opened and kept open until the end of the stroke.

While this method may seem at first thought to be the exact equivalent of admitting gas during the first half of the stroke and then cutting it off, and therefore mere "quality regulation," it is not at all so in practice. When gas is admitted from the beginning of the stroke until some point considerably short of full stroke, say three fourths of the stroke, the air which continues to enter during the remainder of the stroke tends to sweep the gas away from the vicinity of the inlet port and to diffuse it throughout the cylinder contents of air and burned gases. The result is a weak mixture and possibly an absence immediately around the inlet port of any combustible at all. Consequently, if the igniter be located near the inlet port, which is usually the case, the mixture may not be ignited at all, and certainly will not inflame rapidly even if ignited. On the other hand, when the gas is admitted during the latter part of the stroke, there is always a good mixture near the inlet port at the end of the compression stroke and there is much less tendency for the gas to be diffused throughout the whole mass of air and form a poor mixture. Since air is admitted throughout the entire suction stroke the combined volume of air and gas taken in is always the same, as in the case of simple "quality regulation"; consequently, the compression is the same at small loads as at full load.

So far as the author knows, this method has not been applied to gasolene engines, though there is no reason why it should not be applicable when a constant-level vaporizer is not used.

# VIII

## SOME CONSIDERATIONS OF DESIGN

Gas-engine design is yet far from being an "exact science" or anything like an approximation to it. Experience thus far gained, however, has developed some hard-and-fast rules which are of great assistance to young designers. The more important of these follow.

### Cylinder Construction

The inner wall of a cylinder should not be made to transmit axial stresses if the requisite strength necessitates the use of a very thick wall. When the thickness greatly exceeds three inches, the retardation of heat transmission is liable to cause serious troubles, such as deterioration of the inner surface by the prolonged action of excessive heat, unequal expansion of the inner and outer parts of the inner wall, irregular combustion due to ineffective cooling, etc.

The interior of the cylinder should not contain any recesses of great depth relative to their cross section, or pockets which open into the main part of the cylinder through restricted passages. The hot bulb used on some kerosene engines is, of course, an exception to this rule, but the very fact of its applicability to the ignition of the mixture is a forceful demonstration of the objectionable influence of such pockets in cylinders where the mixture is intended to be ignited by other means. They interfere by storing hot gases which pre-ignite the fresh mixture (before compression is complete), and may even cause back-firing, or the ignition of the incoming mixture while the inlet valve is still open.

The contour of the cylinder should be as simple and symmetrical as possible in order to avoid inequalities in expansion when the engine heats up. A perfectly straight cylinder and combustion chamber, with a hemispherical head, is ideal up to the size where the ratio of volumetric capacity to wall surface area in the combustion chamber becomes so large that the maximum quantity of heat per cycle that can be got out of the cylinder is too small to prevent damage to the walls; this construction is not practicable in any but small engines of the single-acting type, but the nearer it can be approached, within the limit mentioned, the fewer difficulties will there be to overcome in making the engine operate satisfactorily under a wide range of conditions.

## Valves and Operating Gear

When natural gas or city illuminating gas is the fuel used, mixing and governing valves may be of almost any form. With producer, coke-oven or blast-furnace gas, any type of sliding valve is likely to cause trouble by reason of the accumulation of soot and dirt on its working surfaces, unless the gas can be cleaned to an extraordinary degree.

The valves opening directly into the cylinder or combustion chamber must be of the poppet or mushroom type because the temperatures encountered are so high that it is impossible to keep a sliding valve tight on its seat without entailing an utterly impractical amount of friction and resultant wear. The stems of poppet valves should be long and guided near both ends in order to seat the valves correctly.

It is advantageous to arrange the admission valves so that air alone is admitted first, followed by the combustible mixture. The initial blast of air, though very brief, sweeps away the hot burned gases remaining in the cylinder from the previous combustion and usually prevents any tendency to ignite the incoming fresh mixture.

Eccentrics have the advantage over cams, for opening the valves, of quieter operation and less shock to the valve-gear shaft and related mechanism. Their employment involves the use also of wipers or their equivalent for moving the valves, and these have the advantage of maximum leverage against the valve at the moment of starting it from its seat, when the resistance of an exhaust valve is greatest. This type of mechanism, however, has more wearing surface than the cam arrangement, and one of them—the eccentric block and strap—is especially crude and difficult to keep in condition.

For transmitting the motion of eccentrics or cams to the valves, either pull rods or bell cranks are preferable to push rods. The last must be made much heavier than either of the first two forms in order to transmit equal force without springing or buckling.

# IX

## CARE AND MANAGEMENT OF ENGINES

In order to handle any internal-combustion engine intelligently the operating engineer must know the elementary principles upon which its working depends. For this reason the preceding chapters have been presented before going into questions of actual operation.

### Starting an Engine

A gas or oil engine is not inherently self-starting, for the reason that there is no reservoir of the working medium at working pressure, as there is in a steam plant. Consequently, such an engine must be started by some means not employed during its regular operation. Small engines are usually started by hand, by turning the flywheel over until a charge of mixture has been drawn in, compressed and ignited; after the first explosion occurs, the engine will continue to run automatically if it is in normal condition. Large engines cannot be turned over by hand, and must, therefore, be started by power. A favorite method is to admit compressed air, by means of special valve gear, to one cylinder of a multicylinder engine or one end of a cylinder of a double-acting engine, driving the engine with the air until it "picks up" the regular charge and fires it. The compressed-air tank is kept filled at the desired pressure—50 to 100 lbs. per square inch—by means of a small compressor driven by the engine when it is in regular operation. Another method, sometimes used in plants containing electric generators, is to provide a small motor arranged to turn the engine over at moderate speed through a gear meshing with a ring of gear teeth either bolted to or cut in the edge of the flywheel rim. The normal running speed of the engine is higher than that at which the motor drives it, and when the engine "picks up" its running cycle a flyball governor operates to unmesh the motor gear and shut down the motor.

Before starting a gas or oil engine five precautions are necessary: the load must be taken off; the ignition system must be inspected and, if necessary, put in proper condition; the lubricating oil feeds must be turned on; the fuel shut-off valve must be opened, and the cooling water must be turned on. It will be found a good plan to attend to these preliminary details in the sequence just given.

If the engine is direct-connected to an electric generator, the load is "taken off" by merely opening the main switch which connects the generator to the circuit. If the load is purely mechanical, a friction clutch must be provided by means of which the engine may be connected to or disconnected from the machinery it drives.

Failure to start promptly when the ignition and fuel valve have had proper attention is usually due to a very abnormal mixture—either much too poor or much too rich. If the ignition system is known (not merely supposed) to be all right in every respect, and the fuel cock is open, the quickest way to get a balky engine started is to adjust the mixture valve for the smallest possible proportion of fuel (or largest possible proportion of air), turn the engine over through one cycle; then increase the proportion of fuel in the mixture and try again, and so on until a mixture is obtained which will fire promptly.

If the time of ignition is adjustable, and it is so on almost all modern engines, it should be set so that the igniter acts immediately *after* the crank has reached the actual dead center; when the engine has "picked up," the ignition should be adjusted ahead sufficiently to produce the best results, as explained in the following section.

## Running an Engine

One of the most deplorable errors of judgment is the assumption that a gas or oil engine requires no attention after it has been started properly. At least as much attention is required as in running a steam engine, and generally more, if reliable, continuous service is expected. If the quality of the fuel is constant, as in the case of an oil engine, or nearly so, as in running on city gas or with a constant load on producer gas, the minimum of attention will be required. But if the quality varies much and frequently, as when running on natural gas or with a varying load on suction producer gas, a good deal of attention is necessary if reliable service and good economy are to be obtained.

In order to obtain the best fuel economy the proportion of fuel to air and the time of ignition must be adjusted with much carefulness. A violent explosion, giving an indicator diagram with a high, sharp peak, is not desirable. It is much better to dilute the mixture more (use a larger proportion of air to fuel) and advance the ignition so as to get prompt firing. This, of course, can be done only by experimentation, and it will facilitate matters to take three or four indicator diagrams after each adjustment. With a large proportion of air to gas or to oil and an increased advance of ignition, an indicator diagram is obtained which does not rise so high at the explosion end, but is much "fatter" from about one third stroke to the toe. By thus manipulating the mixture and ignition it is possible to get a larger diagram area—which means more power—with a given intake of fuel than when the mixture is more nearly perfect and the explosion pressure is higher. This is illustrated by the accompanying table

| Test Group No. | Air to Gas. | Explosion Pressure. | Mean Effective Pressure. | Ignition Advance. | B.t.u. per Horse-power-Hour. |
|---|---|---|---|---|---|
| 1 | 12 | 320 | 86 | 10 % | 15,320 |
| 2 | 14 | 347 | 89 | 13 % | 14,730 |
| 3 | 16 | 363 | 87 | 10 % | 13,270 |
| 4 | 18 | 352 | 90 | 15 % | 11,500 |
| 6 | 20 | 338 | 92 | 18 % | 10,180 |

giving five test averages of a natural-gas engine having a compression pressure of 130 lbs. per square inch, absolute. Each figure in the table is the average of five test figures obtained under as nearly constant conditions as possible. The speeds varied somewhat, of course, and to compensate for this the B.t.u. per horse-power-hour have been reduced to a common speed of 250 revolutions per minute, at which speed the engine was rated.

When the quality of the gas varies rapidly and appreciably, the best that one can do is to adjust the mixture and time of ignition for the average conditions, and in order to do this a fairly steady load must be obtainable. Rig up a wire on the engine frame so that the end of it can be adjusted to indicate the position of the governor reach rod to the throttle or cut-off or whatever the regulating mechanism may be. Cut down the proportion of gas little by little, advancing the ignition each time until the governor is admitting the minimum charge to the cylinder. A point will be found finally where any further decrease in the proportion of gas will cause the governor to begin increasing the charge; that is the point of maximum economy for the conditions under which the engine is running.

The adjustment of the ignition timing to get the best position requires a good deal of experience. It is not usually sufficient to take the knocking of the engine as a guide; while knocking is always an indication of excessive ignition advance (provided less advance stops the knocking), it is not always certain that the advance is not excessive after it has been reduced just enough to stop the knocking. An indicator diagram or a close watch on the governor will serve as a sure check, however. The ignition should not be advanced more than just enough to give the maximum efficiency of combustion under the existing conditions—that is, the maximum area of indicator diagram for a given intake of fuel per cycle.

The cooling water flow does not need to be adjusted with any such nicety as the mixture and the timing of ignition. Theoretically, the jacket water should issue as hot as possible without causing the piston to run so hot as to interfere with lubrication. Practically, however, it makes little difference in economy whether the rise of temperature in the jacket water be 40° or 140° F. For single-acting engines without water-cooled pistons it is usually sufficient to let the temperature of the water rise 80 degrees. For engines which have water-cooled pistons and valves, the *temperature rise* of the cylinder-jacket water should be kept down to about 80 degrees, that of the pistons and valves to about 60 degrees, and that of the cylinder head jackets to not over 100 degrees. Of course, these are general average figures; the builders of an engine will always advise as to the best practice in the matter of cooling their particular engine.

The faster the water flows through the jackets, the more heat it abstracts per minute from the gases in the cylinder, and the less will be the temperature rise of the water. Conversely, a lower rate of flow takes out less heat per minute and produces a greater temperature rise in the water. Therefore, if the temperature at which the water enters the jacket remains constant, the temperature at which it is discharged is a rough guide as to the temperature inside the cylinder—that is, the higher the discharge-water temperature the higher will be the temperature inside the cylinder. The temperature of the jacket-water discharge, therefore, is a useful indication of conditions in the cylinder. If an engine has been running right along with a discharge temperature of 150 degrees, for example, and the temperature suddenly rises to 180 degrees without any change in the rate of flow, something has happened inside the cylinder. Either the load has increased very much or the piston is not being properly lubricated, or some other mishap has started the rings to cutting the cylinder wall. An appreciable increase in the jacket-water discharge temperature, therefore, should be investigated immediately.

Proper lubrication of a gas- or oil-engine cylinder is a problem far more difficult than the lubrication of a steam-engine cylinder. The high temperatures and lack of moisture make it impossible to use the heavy class of oils used in steam cylinders; the oil must be as light and thin as possible without sacrificing its viscosity beyond the permissible limit. Special oils are manufactured expressly for lubricating gas-engine cylinders, and some one of these should be used. An unsuitable oil will carbonize and form crusty deposits on the piston rings, exhaust valve edges and igniter parts. When such deposits are found, one of two things is certain: either the oil is not suitable for the conditions under which the engine works or too much of it has been used. A fairly good guide as to the proper quantity to be used is a sample of the exhaust gases taken out through a ½-in. "bleeder" tapped into the bend of that elbow in the exhaust pipe which is nearest the engine. The end of the "bleeder" which is tapped into the elbow should point squarely toward the out-coming exhaust gas. If the gas discharged by the bleeder is blue, too much cylinder oil is being used; the rate of feed should be very gradually decreased until the exhaust gas has no bluish tinge, *and then decreased one more notch.*

When a vertical engine is built with a closed crank case and designed for splash lubrication of the crank and piston pins, an oil especially prepared for this service should be used in the crank case.

On all other bearings, such as the main shaft, gear shaft, open cranks, and valve mechanism, a good grade of machinery oil such as is used on steam-engine bearings should be applied.

At regular intervals, varying according to the character of the service, but never longer than a year, the engine should be thoroughly inspected, inside and out, and any irregularities corrected.

## Shutting Down

It may seem superfluous to say anything about shutting down an engine—"anyone can shut down." Yes, but not everyone does shut down—properly. A careful engineer always shuts down by cutting off the fuel supply. Then he opens the ignition circuit, if the ignition current is derived from electric battery cells, either primary or storage; in any event, he sets the ignition timer in the retarded, or dead center, position ready for starting. Then the oil feed is stopped and finally the cooling water is cut off and the jackets drained. This last precaution is not always necessary in moderate weather, but in climates where freezing temperatures occur in winter it is a good plan to form the habit of draining the jackets every time an engine is shut down over night or for an indefinite period; then it will not be so easily forgotten when the thermometer is around zero.

## Troubles

The troubles most commonly encountered in running gas and oil engines are (1) back-firing, (2) premature ignition, (3) missing explosions, and (4) after-firing.

*Back-firing* is a premature explosion occurring during the suction stroke, while the inlet valve is open. It is unmistakable, because the explosion is "delivered" through the air-intake pipe, and the noise, if the engine is of considerable size, is deafening. Back-firing may be due to a leaky exhaust valve, which permits hot gases to be sucked back into the cylinder while the fresh charge is being drawn in; or to a mixture so imperfect that it burns throughout the entire exhaust stroke, igniting the incoming mixture just before the exhaust valve closes; or to improper valve setting, by which the exhaust valve is held open too long after the piston has started on the suction stroke.

*Premature ignition* may be due to too high compression for the character of the fuel used, or, to put it reversely, to the use of a fuel too inflammable for the compression of the engine; the remedy is to use a smaller proportion of fuel to air, and if this results in missing explosions, that fuel cannot be used in that engine unless its compression can be reduced. Premature ignition may be due also to carbon deposits on the piston face or on the wall of the combustion chamber, or on the igniter plug. Such deposits are likely to remain incandescent long enough to overheat the incoming charge. Small pockets in the wall of the combustion chamber also cause premature ignition sometimes by retaining hot burned gases until after the suction stroke begins. These deposits and trapped gases do not fire the incoming charge and cause back-firing, as a rule, but they heat the charge to such a degree that it cannot stand the full temperature rise due to the complete compression stroke, because it reaches the firing temperature before that stroke is completed.

*Missing explosions,* or "skipping" in engines governed by any other than the hit-and-miss method, is nearly always due to some defect in the ignition system. When "skipping" occurs, therefore, it is a good plan to go over the ignition system thoroughly to make sure that it is working properly. Some common defects are broken wires, loose primary connections (in a jump-spark system), damaged insulation on connecting wires or parts of the apparatus, and fouled igniter plugs. An excessive accumulation of carbon on the insulation of an igniter plug will afford a leakage path across from one electrode to the other and most or all of the current which should fire the mixture will pass along this path and produce no spark. Excessive feeding of lubricating oil will foul a plug in this way in a comparatively short time. "Skipping" may also be due to a bad mixture or to too light a load. If the ignition system is *known* to be in proper condition, try adjusting the mixing valve one way or the other, the least bit at a time. If the "skipping" is increased by the change, move the handle the opposite way, a trifle beyond where it originally was. A few trials will determine positively whether or not the mixture is to blame. Too light a load may cause "skipping" by reducing the compression of a throttling engine excessively or impoverishing the mixture below the igniting point if quality regulation is employed. If the load cannot be increased, there is no remedy in such a case.

*After-firing* is the explosion in the exhaust passages or muffler of a charge which has failed to ignite in the cylinder. The cause, obviously, is "skipping," and the best remedy is to stop the "skipping." If this cannot be done, the injection of a spray of water into the exhaust pipe near the engine will stop the after-firing. The water can usually be tapped from the circulating system which cools the cylinder barrel or head.

# X

## PRESSURE, TEMPERATURE, AND OUTPUT CALCULATIONS

### Gases

A PERFECT gas, if it existed, would disappear entirely—become reduced to zero volume—if its temperature ~~and pressure~~ were reduced to absolute zero. There is no perfect gas, but it is assumed that air and the gases burned in an engine cylinder behave like perfect gases and that their specific heats are constant at all temperatures. Neither assumption is true, but the discrepancy is not serious enough to make the results of calculations useless, and it is impracticable to make corrections for the lack of accuracy entailed. The following explanations are based on the assumptions mentioned.

In order to raise the temperature of a gas 1° F., a certain quantity of heat must be added to it. If the gas is confined in a rigidly closed vessel so that it cannot expand, the quantity of heat required to raise the temperature of one pound of it one degree is called the "specific heat at constant volume," this quantity is represented by the symbol $C_v$, the $C$ being the initial of the French word "chaleur," which means "heat," and the subscript indicating constant volume.

If the gas be confined in a cylinder closed at one end by a rigid head and in the other direction by a piston, adding heat to it will cause it to expand and push the piston outward against the pressure of the atmosphere. The gas therefore does work, so that the quantity of heat necessary to raise the temperature of one pound of it one degree is greater than when it cannot expand and do work. This quantity—the number of British thermal units required to raise the temperature of one pound of gas one degree when it is free to expand against a constant pressure equal to the initial pressure of the gas—is called the "specific heat at constant pressure," and is represented by the symbol $C_p$.

The arithmetical difference between the specific heats is the quantity of heat required to expand the gas against the constant pressure when its temperature is raised one degree, there being no interference by mechanical friction. Suppose the pressure against which it expands be that of atmosphere (2116.3 lbs. per square foot) and the increase in cubic feet be represented by $V$; then the work done in expanding will be:

$$2116.3\ V \text{ ft. lbs.}$$

In order to enable one pound of gas to do this work, in addition to raising its temperature one degree, heat energy must be added to it to the extent of

$$C_p - C_v = C_e,$$

and since the heat energy in one British thermal unit is equivalent to 778 ft. lbs. of mechanical energy, the mechanical energy added to the gas will be

$$778\ C_e.$$

Therefore

$$2116.3\ V = 778\ C_e,$$

if the heat added be sufficient to raise the temperature one degree and expand the gas against atmospheric pressure. If the pressure be anything else and the temperature elevation be more or less than one degree, and if the weight of gas be $W$ pounds, the case is covered by representing the pressure and temperature rise by symbols, thus:

$$P\ V = 778\ C_e\ t\ W,$$

in which $t =$ the temperature rise in Fahrenheit degrees.

Absolute pressures and temperatures are used in considering the behavior of gases for reasons given in the next paragraph. Absolute pressure is gauge pressure + atmospheric pressure, and is taken at 14.69 lbs. per square inch or 2116.3 lbs. per square foot. Absolute temperature is taken at Fahrenheit thermometer temperature + 460, the absolute zero being 460 degrees below the Fahrenheit thermometer zero. Volumes are in cubic feet throughout this discussion.

All calculations are based on absolute pressures and temperatures because it is considered that a gas in any condition has been expanded to that condition from zero pressure, temperature, and volume. Consequently, no matter what the pressure $P$ and temperature $T$, and the resulting volume $V$, it is considered that the gas has increased its volume from zero to $V$ cubic feet against a constant pressure of $P$ pounds per square foot due to a gain of heat which has also raised the temperature from zero to $T$ degrees, absolute. It follows, therefore, that

$$P\ V = 778\ C_e\ T\ W, \tag{a}$$

no matter what the values of $P$, $V$, $T$ and $W$ may be. This being accepted as a working hypothesis, it is evident that when the condition of a given weight of gas is changed, as by compression, combustion, or expansion under artificial influences, since

$$\frac{P\ V}{T} = 778\ C_e\ W,$$

and $C_e$ and $W$ are constant, the ratio

$$\frac{PV}{T}$$

remains constant through the change in condition. Indicating the condition before the change by the subscript $_1$ and the condition after the change by $_2$,

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2}. \tag{b}$$

### COMPRESSION

The first change to which the gases of an engine are subjected in working through a cycle is that due to compression. Using the subscript $_a$ to indicate the conditions of the cylinder contents just before compression begins and the subscript $_c$ to indicate the conditions when compression is complete,

$$\frac{P_c V_c}{T_c} = \frac{P_a V_a}{T_a}.$$

From this fundamental equation are derived the formulas:

$$P_c = P_a r_c^n, \qquad \text{(c)*}$$

and

$$T_c = T_a r_c^{n-1}, \qquad \text{(d)*}$$

in which $r_c$ is the compression ratio, as explained on page 18.

---

* *Derivation of Equations* (c) *and* (d):

$$\frac{P_c V_c}{T_c} = \frac{P_a V_a}{T_a},$$

consequently,

$$\frac{P_c}{P_a} = \frac{T_c}{T_a} \cdot \frac{V_a}{V_c}.$$

Now, with adiabatic compression,

$$\frac{T_c}{T_a} = \left(\frac{V_a}{V_c}\right)^{k-1},$$

in which

$$k = \frac{C_p}{C_v};$$

therefore

$$\frac{P_c}{P_a} = \left(\frac{V_a}{V_c}\right).$$

Compression, however, is not adiabatic; heat is lost to the jacket, so that an exponent of smaller value than $k$ must be used; this is usually represented by $n$. Making this change

$$\frac{T_c}{T_a} = \left(\frac{V_a}{V_c}\right)^{n-1} = r_c^{n-1},$$

and

$$\frac{P_c}{P_a} = \left(\frac{V_a}{V_c}\right)^{n} = r_c^{n}.$$

These obviously transpose into

$$T_c = T_a r_c^{n-1}. \tag{d}$$

and

$$P_c = P_a r_c^n. \tag{c}$$

If combustion of the mixture were complete and instantaneous while the piston was stationary at the end of the compression stroke, and if there were no loss of heat to the water jacket, the rise of temperature produced by combustion would be given by the formula

$$\frac{H}{C_v} = T_x - T_c,$$

in which

$H$ = B.t.u. evolved by combustion, per pound of cylinder contents,
$C_v$ = Specific heat of cylinder contents at constant volume,
$T_x$ = Maximum temperature attained by cylinder contents.

But complete and instantaneous combustion is out of the question practically, as already explained in the earlier chapter on pressures and temperatures, and heat must be taken away by the jacket water in order to keep the cylinder from becoming red-hot. It is impossible to predict just how much these imperfections will reduce the temperature rise, but by using a symbol to represent the proportion of the ideal temperature rise actually obtained, a definite formula may be deduced for the actual temperature rise, thus:

$$\frac{H u}{C_v} = T_x - T_c, \qquad \text{(e)}$$

in which $u$ represents the ratio of actual temperature rise to that which would be obtained with perfect and instantaneous combustion and no heat loss. Thus, if $\frac{H}{C_v}$ were equal to 3000 and the actual rise of temperature were 1620, the value of $u$ would be 0.54. As a general rule, with gases of high heat value the value of $u$ is lower than with gases of low heat value, if the proportion of air to gas be equally favorable in both cases. One reason for this is that engines using rich gases cannot work with high compression because of the liability to premature ignition; consequently, their clearances are larger than those of engines which burn producer or blast-furnace gases, and the cooling surfaces are larger. This allows a relatively greater loss of heat during combustion and lowers the efficiency of the process, thereby reducing the value of $u$, which covers heat loss as well as slowness of combustion. Moreover, it is generally the case that a mixture formed with rich gases is neither as well-proportioned nor as thoroughly compounded as one made with lean gases, and this causes relatively sluggish combustion.

With oil fuels the value of $u$ is usually much higher than with gaseous fuels. Whether or not this is because oil atoms are more easily brought into contact with their proportion of oxygen than gas atoms the author does not know, but it is certain that the rise of pressure is relatively much more violent with gasolene than with any gaseous fuel. Explosion pressures of 350 to 400 lbs. are often obtained with compression pressures of 80 to 90 lbs. per square inch, using gasolene, while with illuminating and natural gases compression pressures of 100 to 125 lbs. are usually accompanied by explosion pressures of 250 to 350 lbs.

The cylinder contents in a nonscavenging engine (few engines actually clear out all of the burned gases) consist of a fresh mixture of air and gas or air and oil vapor and the burned gases remaining over from the preceding explosion. The volume of the fresh mixture in a four-stroke-cycle engine is usually equal, before compression, to the piston displacement during the suction stroke because the clearance space remains filled with burned gases at the end of the expulsion stroke. Consequently the heat per pound of cylinder contents will be the total heat in the fresh mixture divided by the weight of the cylinder contents. The total heat in the mixture is practically equal to

$$\frac{h}{1+a} \times V_s;$$

consequently, the number of heat units per pound of cylinder contents is

$$\frac{h}{1+a} \times \frac{V_s}{W} = H, \tag{f}$$

in which

$h$ = B.t.u. per cubic foot of gas alone at the temperature of the mixture immediately before compression,
$a$ = Cubic feet of air per cubic foot of gas in the fresh mixture,
$V_s$ = Cubic feet of piston displacement,
$W$ = Pounds of cylinder contents.

Substituting this equivalent for $H$ in equation (e), transposing and making the substitutions explained in the footnote, the result is the formula:

$$T_x = T_c + K_g \left( 1 - \frac{1}{r_c} \right) \frac{T_a}{P_a}. \qquad \text{(g)*}$$

in which

$$K_g = \frac{778\ (k-1)\ h\ u}{1+a} \tag{h}$$

---

* *Derivation of Equation (g):*

The heat units per pound of cylinder contents are given by the equation

$$\frac{h}{1+a} \times \frac{V_s}{W} = H; \tag{f}$$

substituting this equivalent for $H$ in equation (e) gives the equation

$$\frac{h\ u\ V_s}{C_v\ (1+a)\ W} = T_x - T_c,$$

In this last equation

$$k = \frac{C_p}{C_v}.$$

The relation between compression and explosion temperatures and pressures is

$$\frac{P_x}{P_c} = \frac{T_x}{T_c}.$$

---

which transposes to

$$T_x = T_c + \frac{h\ u\ V_s}{C_v\ (1 + a)\ W}. \qquad \textbf{(t)}$$

Equation (a) furnishes an equivalent for the product of $C_v\ W$ which is easier to evaluate in practice. Thus, equation (a) may be transposed into

$$W = \frac{P_a\ V_a}{778\ T_a\ C_s},$$

and

$$C_s = C_p - C_v = C_v\left(\frac{C_p - C_v}{C_v}\right),$$

which reduces to

$$C_s = C_v\left(\frac{C_p}{C_v} - 1\right),$$

or

$$C_s = C_v\ (k-1).$$

Substituting this equivalent for $C_s$ and transposing $C_v$ to get it with the $W$, produces the equation

$$C_v\ W = \frac{P_a\ V_a}{778\ T_a\ (k - 1)}.$$

Now substituting

$$\frac{P_a\ V_a}{778\ T_a\ (k - 1)}$$

in the place of $C_v\ W$ in formula (t) above, for explosion temperature, gives:

$$T_x = T_c + \left(\frac{h\ u\ 778\ (k-1)}{1 + a} \times \frac{T_a}{P_a} \times \frac{V_s}{V_a}\right).$$

This may be made more manageable by substituting the equivalent of $V_s \div V_a$ in terms of the compression ratio. Thus:

$$\frac{V_s}{V_a} = \frac{V_a - V_c}{V_a} = 1 - \frac{V_c}{V_a}.$$

But

$$\frac{V_c}{V_a} = \frac{1}{r_c},$$

so that

$$\frac{V_s}{V_a} = 1 - \frac{1}{r_c}.$$

Making this final substitution gives, for the explosion temperature formula,

$$T_x = T_c + \frac{778\ (k - 1)\ h\ u\ T_a\left(1 - \frac{1}{r_c}\right)}{(1 + a)\ P_a},$$

which is identical with equation (g) when equation (h) is applied.

Accepting the equivalent of $T_x$ given by equation (g) and making the transpositions and substitutions explained in the footnote, the following formula for maximum pressure of combustion is obtained:

$$P_x = P_c + K_g\,(r_c - 1). \qquad \text{(i)*}$$

So far as it affects the pressure and temperature at the moment of release, the expansion ratio is

$$r_e = \frac{V_e}{V_x};$$

and since $V_x = V_c$,

$$r_e = \frac{V_e}{V_c}.$$

In a two-stroke engine $r_e$ is practically equal to $r_c$; in a four-stroke engine the relation between the two, so far as it affects the release pressure and temperature, is

$$r_e = f\,(r_c - 1) + 1,$$

in which $f$ is the fraction of the piston stroke that is completed when the exhaust valve opens. Thus, if the exhaust valve opened at 90 per cent.

---

* *Derivation of Equation* (*i*):

Applying formula (b) to the pressure increase of the combustion process, and representing the maximum pressure of combustion or explosion by $P_x$,

$$\frac{P_x\,V_x}{T_x} = \frac{P_c\,V_c}{T_c},$$

and as the volume is practically constant, $V_x = V_c$. Consequently,

$$\frac{P_x}{T_x} = \frac{P_c}{T_c},$$

and this transposes to

$$P_x = \frac{P_c\,T_x}{T_c}.$$

Accepting the equivalent of $T_x$ given by equation (g), dividing it by $T_c$ and multiplying by $P_c$ gives the following:

$$P_x = P_c + \frac{K_g\,T_a\left(1 - \frac{1}{r_c}\right)P_c}{P_a\,T_c}.$$

But, according to equation (b),

$$\frac{P_c}{P_a} \times \frac{T_a}{T_c} = \frac{V_a}{V_c} = r_c,$$

so that $r_c$ can be put in the numerator of the fraction and the two temperatures and pressures eliminated.

And since $1 - \frac{1}{r_c}$ is already there and

$$r_c \times \left(1 - \frac{1}{r_c}\right) = r_c - 1,$$

the formula becomes:

$$P_x = P_c + K_g\,(r_c - 1). \qquad \text{(i)}$$

The student should not confuse $k$ with the exponent $n$, but remember that $k = C_p \div C_v$ always, while $n$ is always less than that.

of the expansion stroke and the compression ratio were 4, the expansion ratio would be:

$$0.9\ (4 - 1) + 1 = 3.7.$$

The pressure at the moment of exhaust opening is

$$P_e = P_x \div r_e{}^n, \tag{j}$$

and the gas temperature at the same moment is

$$T_e = T_x \div r_e{}^{n-1}. \tag{k}$$

Tables 4 and 5 give the release pressures and temperatures corresponding to several usual explosion pressures and temperatures, assuming expansion exponents of 1.29 and 1.32; these are common practical values, the former for double-acting or other thoroughly cooled engines and the latter for single-acting or other moderately cooled engines as they are usually operated.

### Work Done Per Cycle

The work done in compressing the cylinder contents is

$$\frac{P_c\ V_c - P_a\ V_a}{n_c - 1} = ft.\text{-}lbs._c,$$

and the work done by the gases on the piston during expansion is

$$\frac{P_x\ V_c - P_e\ V_e}{n_e - 1} = ft.\text{-}lbs._e.$$

In this case it is not fair to the four-stroke-cycle engine to consider the point at which the exhaust valve begins to open as the end of the expansion stroke, because, as explained in connection with Fig. 3, the pressure does not drop at once to atmospheric. It is more nearly accurate to assume that the valve opens at a point about halfway between the point of actual opening and the end of the piston travel. For most of the indicator diagrams which the author has measured and checked up, the assumption that the exhaust valve opened at a point of the piston stroke four tenths of the remaining travel beyond the point of actual opening, and then reduced the pressure instantaneously to atmosphere, gave calculated diagram areas almost exactly agreeing with the actual diagram areas. Table 7 has been computed on this basis, the formula being

$$r'_e = r_c - 0.6\ (1 - f)\ (r_c - 1). \tag{l}$$

The volume $V_e$, therefore, is that volume and the pressure $P_e$ is that pressure which would exist behind the piston at the moment of release if the actions were those described above; that is,

$$V_e = V_c \times r'_e,$$

in which $r'_e$ is the effective expansion ratio as given by equation (1) or Table 7.

The net work done on the piston is the difference between the work of expansion and the work of compression, thus:

$$Ft.\text{-}lbs._e - ft.\text{-}lbs._c = net\ indicated\ ft.\text{-}lbs.\ per\ cycle.$$

From this fundamental equation is derived the more conveniently applicable equation:

$$(P_x K_e - P_c K_c)\frac{V_s}{r_c - 1} = ind.\ ft.\text{-}lbs.\ per\ cycle. \qquad \text{(m)*}$$

in which

$$K_e = \frac{1 - \dfrac{1}{r_c^{\,n-1}}}{n_e - 1},$$

and

$$K_c = \frac{1 - \dfrac{1}{r_c^{\,n-1}}}{n_c - 1}.$$

---

* *Derivation of Equation (m):*

The net work done on the piston per cycle is

$$\frac{P_x V_c - P_e V_e}{n_e - 1} - \frac{P_c V_c - P_a V_a}{n_c - 1} = ft.\text{-}lbs.\ per\ cycle.$$

and since $V_e = V_c r'_e$ and $V_a = V_c r_c$,

$$\left(\frac{P_x - P_e r'_e}{n_e - 1} - \frac{P_c - P_a r_c}{n_c - 1}\right) \times V_c = ft.\text{-}lbs.\ per\ cycle.$$

Now,

$$P_e r'_e = \frac{P_x}{r'_e{}^{n-1}},$$

and

$$P_a r_c = \frac{P_c}{r_c^{\,n-1}};$$

consequently

$$P_x - P_e r'_e = P_x\left(1 - \frac{1}{r'_e{}^{n-1}}\right),$$

and, correspondingly,

$$P_c - P_a r_c = P_c\left(1 - \frac{1}{r_c^{\,n-1}}\right).$$

Moreover,

$$V_c = \frac{V_s}{r_c - 1}$$

Substituting these three equivalents, the result is

$$\left[P_x\left(\frac{1 - \dfrac{1}{r_e^{\,n-1}}}{n_e - 1}\right) - P_c\left(\frac{1 - \dfrac{1}{r_c^{\,n-1}}}{n_c - 1}\right)\right] \times \frac{V_s}{r_c - 1} = ind.\ foot\text{-}pounds\ per\ cycle.$$

which becomes identical with equation (m) upon substituting $K_e$ and $K_c$ for the bracketed fractions.

The numerical values of $K_c$ and $K_e$ are so nearly equal in practice that, in view of the other uncertainties involved in output calculations, it is sufficient to consider them equal. For example, if

$$r_c = 5,\ n_c = 1.35,\ r'_e = 4.85, \text{ and } n_e = 1.32,$$

then

$$K_c = 1.23044,$$

and

$$K_e = 1.23966,$$

the difference being less than one per cent. of the smaller of the two values. Sometimes the value of $K_c$ is larger than that of $K_e$; for example, if

$$r_c = 5,\ n_c = 1.3,\ r'_e = 4.75, \text{ and } n_e = 1.27,$$

then

$$K_c = 1.2767,$$

while

$$K_e = 1.2718.$$

However, as will be explained further along, these constants have to be modified to fit actual engine performances, so that the small difference between their values is of no importance. The theoretical equation for indicated work per cycle may just as well be condensed into

$$\frac{K_c\,(P_x - P_c)\,V_s}{r_c - 1} = \textit{ind. ft.-lbs. per cycle.}$$

Inspection of equation (i) shows that

$$\frac{P_x - P_c}{r_c - 1} = K_g,$$

and substituting this equivalent in the work formula gives as the final theoretical equation:

$$K_g\,K_c\,V_s = \textit{ind. ft.-lbs. per cycle.} \qquad \text{(n)}$$

## Indicated Horse Power

The horse power of any engine is equal to

$$\frac{\textit{Foot-pounds per minute}}{33{,}000}$$

and the number of useful foot-pounds per minute developed in a gas-engine cylinder is equal to the foot-pounds per cycle $\times$ number of explosions per minute. Consequently, if the number of explosions per minute be represented by $N_x$, the indicated power of an engine will be, theoretically,

$$\frac{K_g\,K_c\,V_s\,N_x}{33{,}000} = I.H.P.$$

The relation of this equation to the old steam-engine formula based on mean effective pressure will be obvious upon analysis. The mean effective pressure per *square foot* is equal to the foot-pounds per cycle divided by the piston displacement, so that the mean effective per *square inch* is, theoretically,

$$p_m = \frac{K_g K_c}{144}.$$

The piston displacement in cubic feet is equal to piston area $\times$ stroke, in foot measure, so that

$$L\ A = 144\ V_s,$$

and it will be found that if these equivalents be substituted in the steam-engine formula

$$\frac{p_m\ L\ A\ N_x}{33{,}000} = I.H.P.,$$

the result will be the gas engine equation above.

In all of the equations for pressures, temperatures and work obtained after ignition is accomplished, the factor $K_g$ necessarily appears, because it includes all of the elemental factors which affect the rise of pressure and temperature above the compression point. Referring again to equation (g), it will be found that $K_g$ is the product of $\frac{C_p}{C_v} - 1$, the mixture heat value $\frac{h}{1+a}$, and the "utilization" factor $u$, all multiplied by 778 to reduce heat to foot-pounds. The values of $C_p$ and $C_v$ in this case are those of the entire cylinder contents, consisting of fresh mixture and burned gases remaining in the clearance from the previous explosion. The calculation of these values is very tedious and, moreover, it is difficult to determine the data required to make the computation. The calculation of the heat value ($h$) of the gas is also tedious and the proper value to assign to $a$ (cubic feet of air per cubic foot of gas) is a matter largely of guesswork. As a matter of fact, the ratio of air to gas actually obtained in an engine cylinder is almost impossible of accurate determination; the author has never seen a set of test records which gave any evidence that the air ratio had been correctly ascertained.

## Practical Output Estimation

In view of the foregoing difficulties, the most practical method of estimating engine output is either to base the estimate on the rise of pressure which experience has shown can be obtained reliably with the given fuel and operating conditions (compression pressure, cooling water heat waste, shape of combustion chamber, timing of ignition, etc.), substituting

$$\frac{P_x - P_c}{r_c - 1}$$

for $K_g$ in the equations, or else to assume values for $k$, $u$ and $\frac{h}{1+a}$ which experience has shown to be readily obtainable. In order to facilitate the application of the former of the two methods, Tables 3-A to 3-D have been prepared, the values therein given being based on a large number of indicator diagram analyses. In selecting these values the average values shown by indicator diagrams were not taken, because in rating a gas or oil engine the benefit of all doubt must be accorded to the engine. The values in the table, therefore, will be found somewhat *lower* than one should be able to obtain under average conditions.

In order to apply the second method of estimating, Tables 8, 9 and 10 are provided. From Table 8 one can readily compute the properties of any fuel mixture if the analysis of the fuel is known. In making such computations the results will more nearly agree with practice if the quantity of air per unit of gas be taken at 20 per cent. above the theoretical requirement for oil fuels; 30 per cent. for natural gas; 40 per cent. for less rich gases, such as coke-oven gas, oil-water gas, and ordinary illuminating gas; and 50 per cent. for producer and blast-furnace gases. These percentages are not all in accord with theoretical reasoning, but they seem to be justified by practical experience. Table 9 has been computed on this basis, using some representative fuel analyses. The heat values in this table and Table 8 will seem low to anyone accustomed to heat values based on 32° F., or 0° C.; the explanation will be found in the subheadings of the tables—the figures are for gases at 700° absolute temperature (240° F.) and 13½ lbs. per square inch absolute pressure. The temperature and pressure have been selected as coming nearest to average conditions. In a well-cooled cylinder, with a cooled piston and cooled exhaust valves, the initial mixture temperature may readily be as low as 630° or 640° absolute, but in a cylinder moderately well cooled and having solid valves, the temperature may easily be 800°, especially if the inlet and exhaust ports are near each other.

The next stumbling-block in estimating the power of an engine is the fact that, even if the actual values of all of the other factors were known beforehand, the use of the factor $K_c$ would rarely give correct results. The explanation of this is simple: the compression and expansion curves do not follow strictly the exponential law which is applied to them for lack of something better. Even when the ratio of compression to precompression pressures agrees with equation (c) and the ratio of explosion to release pressures conforms to equation (j), the indicator diagram will usually be from ten to forty per cent. larger in area than called for by either equation (m) or equation (n). This is because the compression curve is slightly *more* concave and the expansion curve very much *less* concave than are the corresponding curves plotted by the formulas. The discrepancy between

## Table 8

## Properties of Gas Engine Fuel Mixture Constituents

ONE CUBIC FOOT, AT 700 DEGREES TEMPERATURE AND 1,944 POUNDS PRESSURE.

Note:—1,944 lbs. per sq. ft. = 13½ lbs. per sq. in.

| Constituent | Weight | B. t. u. | B. t u. to raise temp. one degree Fahr. | | Cubic feet of air utilized in combustion |
|---|---|---|---|---|---|
| | | | Constant pressure | Constant volume | |
| Hydrogen........ | 0.003602 | 186.62 | 0.0122807 | 0.0086901 | 2.41 |
| Methane.......... | 0.02874 | 620.55 | 0.0170393 | 0.0134584 | 9.61 |
| Ethylene.......... | 0.05034 | 998.45 | 0.0203373 | 0.0087088 | 14.42 |
| Carb. oxide....... | 0.05032 | 221.18 | 0.0124742 | 0.0088462 | 2.40 |
| Carb. dioxide..... | 0.07907 | .......... | 0.0171582 | 0.0121373 | ..... |
| Nitrogen......... | 0.05047 | .......... | 0.0123063 | 0.0087174 | ..... |
| Oxygen.......... | 0.05750 | .......... | 0.0125079 | 0.0089194 | ..... |
| Air.............. | 0.05204 | .......... | 0.0123589 | 0.0087891 | ..... |

## Table 9. Properties of Some Representative Fuels and Mixtures

AT 700 DEGREES ABSOLUTE TEMPERATURE AND 1,944 POUNDS ABSOLUTE PRESSURE.

Note:—1,944 lbs. per sq. ft. = 13½ lbs. per sq. in.

| Fuel | B.t.u. per cu. ft | Air Required for Combustion | | Practical Mixtures | | | | |
|---|---|---|---|---|---|---|---|---|
| | | Theoretically | Practically | B.t.u. per cu. ft. | $\frac{Cp}{Cv}=k$ | $\frac{(k-1)\,h}{1+a}$ | $\frac{778\,(k-1)\,h}{1+a}$ | $\frac{778\,(k-1)\,h}{33000\,(1+a)}$ |
| | | Vols. | Vols. | | | | | |
| Gasolene vapor...... | 750 | 14 | 17 | 41.7 | 1.4050 | 16.8885 | 13,139 | 0.3982 |
| Natural gas.......... | 600 | 12 | 14 | 40.0 | 1.3980 | 15.9200 | 12,386 | 0.3753 |
| Oil water gas......... | 353.7 | 5.2 | 8 | 39.3 | 1.4060 | 15.9558 | 12,414 | 0.3762 |
| Dowson producer gas. | 96.7 | 1.09 | 1.6 | 37.2 | 1.4098 | 15.2445 | 11,860 | 0.3954 |
| Mond " " | 93.7 | 1.12 | 1.7 | 34.7 | 1.4085 | 14.1750 | 11,028 | 0.3342 |
| Taylor " " | 91.6 | 1.08 | 1.6 | 35.2 | 1.4083 | 14.3722 | 11,182 | 0.3388 |
| Avg. suc. " " | 90.6 | 1.03 | 1.6 | 35.0 | 1.4095 | 14.3325 | 11,151 | 0.3379 |
| Blast furnace " | 62.4 | 0.68 | 1.0 | 31.2 | 1.4095 | 12.7760 | 9,940 | 0.3012 |

ANALYSES OF THE ABOVE FUELS BY VOLUME.

| Constituent | Gasolene by weight | Natural gas | Oil gas | Dowson gas | Mond gas | Taylor gas | Avg. suc. gas | Blast F'rn'ce gas |
|---|---|---|---|---|---|---|---|---|
| Carbon........... | 85 % | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| Hydrogen......... | 14½% | 2.2% | 51.8% | 18.5% | 22.0% | 17.0% | 14.0% | 2.6% |
| Methane.......... | ...... | 92.6% | 33.0% | 0.5% | 2.4% | 2.5% | 0.8% | ...... |
| Ethylene.......... | ...... | 0.4% | 4.2% | ..... | ..... | ..... | ..... | ..... |
| Carb. oxide....... | ...... | 0.5% | 4.2% | 25.6% | 16.0% | 19.0% | 26.0% | 26.0% |
| Carb. dioxide..... | ...... | 0.3% | 1.4% | 8.0% | 12.0% | 8.4% | 5.0% | 9.4% |
| Nitrogen.......... | ...... | 3.7% | 5.2% | 47.0% | 47.0% | 52.5% | 54.0% | 62.0% |
| Oxygen........... | ...... | 0.3% | 0.2% | 0.4% | 0.6% | 0.6% | 0.2% | 0.0% |

## Table 10. Average Practical Values of $K_g$

$$Kg = 778\ \frac{h}{1+a}\ u\ (k\text{-}1)$$

| Fuel: | Gasolene | Kerosene | Nat'l Gas | Ill'g Gas | Prod. Gas | Blast Gas |
|---|---|---|---|---|---|---|
| Avg. *Kg*: | 9300 | 8100 | 7 x *h'* | 12 x *h'* | 40 x *h'* | 50 x *h'* |

NOTE.—*h'* = B.t.u. per cubic foot of gas at 32° Fahr.

the calculated and the actual curves is due chiefly to the use of formulas based on the assumption that heat is added and withdrawn in a regular manner during expansion and compression, respectively, whereas the addition and withdrawal are far from regular in most cases. The heat loss during the first two thirds of the compression stroke is more rapid, relatively, than during the remaining third, because of the constantly decreasing wall surface through which the heat escapes to the jacket water. The heat interchange during the expansion stroke is more complex. In most cases, the curve starts down about with the calculated curve; a little later it is flattened by an access of heat either from the walls or by after-burning

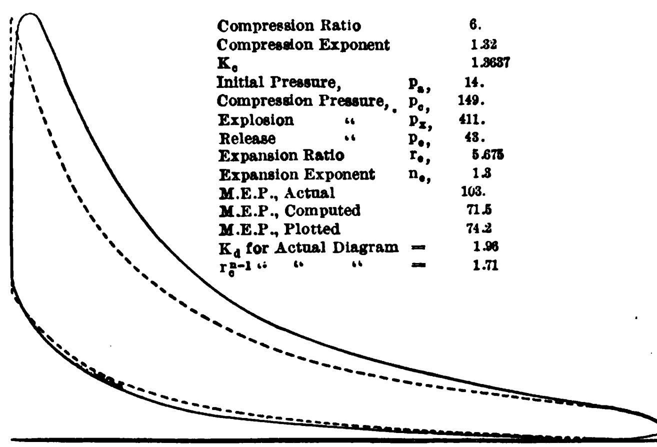



FIG. 40. — ILLUSTRATING DISCREPANCIES BETWEEN ACTUAL AND THEORETICAL INDICATOR DIAGRAMS.

of the gases. No matter what the cause is, there is obviously an irregular accession of heat which raises the curve after expansion begins.

These discrepancies are illustrated in Fig. 40, in which the solid curves are reproduced from those traced by an indicator and the dotted curves were plotted by the formulas (c) and (j), using the pressures shown by the actual indicator diagram and the exponents corresponding to the pressure ratios. Another discrepancy lies in the fact that the "peaks" of nearly all normal indicator diagrams are considerably rounded, whereas the power and work formulas are based on the assumption of a sharp point. If the "peak" representing the maximum explosion pressure were carried up, in the calculated diagram, to the point that it would reach with instantaneous but not adiabatic combustion, and the expansion line were started from that point, the area of the plotted diagram would doubtless be nearer to equality with that of the actual indicator diagram, and equations containing or depending on the factor $K_c$ would approximate more uniformly to accuracy. But it is impossible to predict just where the peak would go with instantaneous combustion, because neither the liberation nor the loss of heat during the explosion can be estimated with any confidence.

In view of these discrepancies between the formulas and the actual performance, it is more practical to substitute a factor for $K_c$ in equation (n) which will vary generally with the compression ratio and yet allow for the absence of regularity in the compression and expansion curves and the rounding of the maximum-pressure point. Representing the substitute factor by $K_d$, the formula for mean effective pressure becomes

$$p_m = \frac{K_g \, K_d}{144}, \tag{p}$$

and this may also be written

$$p_m = \frac{K_d \, (p_x - p_c)}{r_c - 1}. \tag{p2}$$

The indicated horse-power formula, based on mean effective pressure, is

$$\frac{144 \, p_m \, V_s \, N_x}{33{,}000} = I.H.P., \tag{q}$$

and if one prefers to go direct from foot-pounds per cycle to horse power,

$$\frac{K_g \, K_d \, V_s \, N_x}{33{,}000} = I.H.P., \tag{q2}$$

It now remains to deduce a rational value for the empirical factor $K_d$ which is supposed to include $K_c$ and also to make allowance for the discrepancies just discussed. From an analysis of nearly one hundred indicator diagrams, taken under widely varying conditions and from all classes of engines, the author has come to the conclusion that no factor can be worked out that will give calculated results corresponding to even the average results obtained in practice under widely different conditions. The value of $r_c^{\,n-1}$ comes nearer the general average than any other factor that has been tried, and in many cases it has given surprisingly close results.

Tables 11 and 12 will serve to illustrate the differences between actual and estimated engine performances, the estimated results being based on Tables 3-A and 3-C, Table 10, equation (p) and the use of $r_c^{\,n-1}$ for the value of $K_d$. The data in Table 11 were obtained by tests and measurements of the various engines and fuels. No measurements were made of the proportion of air to fuel, the mixture in all cases being adjusted until the best results were obtained in the engine cylinder. Engine No. 5 was a tandem single-acting, No. 6 was a twin-cylinder single-acting, and No. 7 was one of the mammoth twin tandem double-acting engines in California, described in *Power* of January 14, 1908. All of the others were simple single-acting engines.

## Table 11. Seven Test Records

| Test No. | Bore and Stroke | R.p.m. | $N_x$ | $r_c$ | $p_c$ | $p_x$ | $p_m$ | I.H.P.† | Fuel | B.t.u. per cu. ft. cold |
|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 5½ x 9 | 270 | 133 | 3.8 | 75 | 250 | 89.5 | 6.44 | Gasolene | ... |
| 2 | 8.27 x 16.51 | 218 | 98.9* | 6.4 | 149 | 341 | 56.5 | 15.85 | Prod. gas | 127 |
| 3 | 8½ x 13 | 200 | 92* | 4.75 | 113 | 287 | 76.9 | 13.50 | City gas | 585 |
| 4 | 9 x 15 | 202 | 88.9* | 5.5 | 120 | 433 | 107.35 | 23.00 | " | 650 |
| 5 | 9¾ x 17½ | 180 | 90 | 4.3 | 86.7 | 362.7 | 87.4 | 25.95 | " | 600 |
| 6 | 10 x 19 | 190 | 95 | 4.3 | 85.3 | 312.3 | 86.6 | 31.00 | " | 660 |
| 7 | 42 x 60 | 88 | 44 | 4.6 | 86 | 256 | 83.1 | 767.50 | " | 625 |

* Hit-and-miss regulation.
† Power developed in one end of a cylinder in double-acting engines.

## Table 12. Comparisons of Test Results with Estimates

| Test No. | Pressure rise $p_x - p_c$ | | $K_g = \frac{144(p_x - p_c)}{r_c - 1}$ | | $K_d = \frac{144 p_m}{K_g}$ | | Compress'n exponent $n_c$ | | $p_m = \frac{K_g K_d}{144}$ | |
|---|---|---|---|---|---|---|---|---|---|---|
| | Test | Est. | Test | Est. | Test | Est.* | Test | Est. | Test | Est. |
| 1 | 175 | 182 | 9103 | 9300 | 1.432 | 1.493 | 1.32 | 1.30 | 89.5 | 96.42 |
| 2 | 192 | 205 | 5120 | 5080 | 1.589 | 1.745 | 1.28 | 1.30 | 56.5 | 61.56 |
| 3 | 174 | 183 | 6682 | 6780 | 1.657 | 1.698 | 1.34 | 1.34 | 76.9 | 79.95 |
| 4 | 313 | 236 | 10016 | 7800 | 1.543 | 1.668 | 1.30 | 1.30 | 107.35 | 90.35 |
| 5 | 276 | 148 | 12043 | 7200 | 1.045 | 1.549 | 1.29 | 1.30 | 87.4 | 77.45 |
| 6 | 227 | 163 | 9905 | 7920 | 1.259 | 1.549 | 1.28 | 1.30 | 86.6 | 85.20 |
| 7 | 170 | 169 | 6800 | 7500 | 1.760 | 1.630 | 1.30 | 1.32 | 83.1 | 84.90 |

* Assumed to be equal to $r_c^{n-1}$.

## Table 13. Horse-power Constants

FOR DIFFERENT MEAN EFFECTIVE PRESSURES

| M.E.P. | H. P. Constant | M.E.P. | H. P. Constant | M.E.P. | H. P. Constant |
|---|---|---|---|---|---|
| 50 | 0.2182 | 70 | 0.3055 | 90 | 0.3927 |
| 51 | 0.2225 | 71 | 0.3098 | 91 | 0.3971 |
| 52 | 0.2269 | 72 | 0.3142 | 92 | 0.4015 |
| 53 | 0.2313 | 73 | 0.3185 | 93 | 0.4058 |
| 54 | 0.2356 | 74 | 0.3229 | 94 | 0.4102 |
| 55 | 0.2400 | 75 | 0.3273 | 95 | 0.4145 |
| 56 | 0.2444 | 76 | 0.3316 | 96 | 0.4189 |
| 57 | 0.2487 | 77 | 0.3360 | 97 | 0.4233 |
| 58 | 0.2531 | 78 | 0.3404 | 98 | 0.4276 |
| 59 | 0.2575 | 79 | 0.3447 | 99 | 0.4320 |
| 60 | 0.2618 | 80 | 0.3491 | 100 | 0.4364 |
| 61 | 0.2662 | 81 | 0.3535 | 101 | 0.4407 |
| 62 | 0.2705 | 82 | 0.3578 | 102 | 0.4451 |
| 63 | 0.2749 | 83 | 0.3622 | 103 | 0.4495 |
| 64 | 0.2793 | 84 | 0.3665 | 104 | 0.4538 |
| 65 | 0.2836 | 85 | 0.3709 | 105 | 0.4582 |
| 66 | 0.2880 | 86 | 0.3753 | 106 | 0.4625 |
| 67 | 0.2924 | 87 | 0.3796 | 107 | 0.4669 |
| 68 | 0.2967 | 88 | 0.3840 | 108 | 0.4713 |
| 69 | 0.3011 | 89 | 0.3884 | 110 | 0.4800 |

Constant × Piston Displacement in Cu. Ft. × Explosions per minute = I.H.P.

It will be noticed by reference to Table 12 that the empirical values of $K_d$ average up fairly close to the actual values derived from the tests, but the comparisons in Tests 2, 5, and 6 indicate the futility of trying to derive a factor that will approximate accuracy in every case.

When one has a series of indicator diagrams the indicated horse power is, of course, readily computed by means of equation (q). In order to facilitate this computation, Table 13 is provided. The "horse-power constants" are merely the mean effective pressures multiplied by 144 and divided by 33,000. Having the mean effective pressure from the diagram, the corresponding constant is taken from the table, either direct or by extrapolation. Thus, if the mean effective pressure were 76.7 lbs., the factor would be 0.3316 + 0.7 of the difference between this and 0.3360, or

$$0.3316 + 0.00308 = 0.33468.$$

Tables 14-A and 14-B give piston displacements in cubic feet for diameters of 3 to 60 ins. and stroke lengths of $\frac{1}{4}$ in. to 9 ins.

Tables 15 and 16 have been prepared for the convenience of those dealing with engines which depart from the usual pressure and temperature ratios so far as to escape the application of Tables 1, 2, 4, and 5. Table 16 also supplies values of $K_d$ for making rough estimates by means of formulas (p), (p2), and (q2).

## Efficiency

The indicated conversion efficiency is the net work done on the piston per cycle divided by the work equivalent of the heat liberated by combustion per cycle. Expressed as a formula:

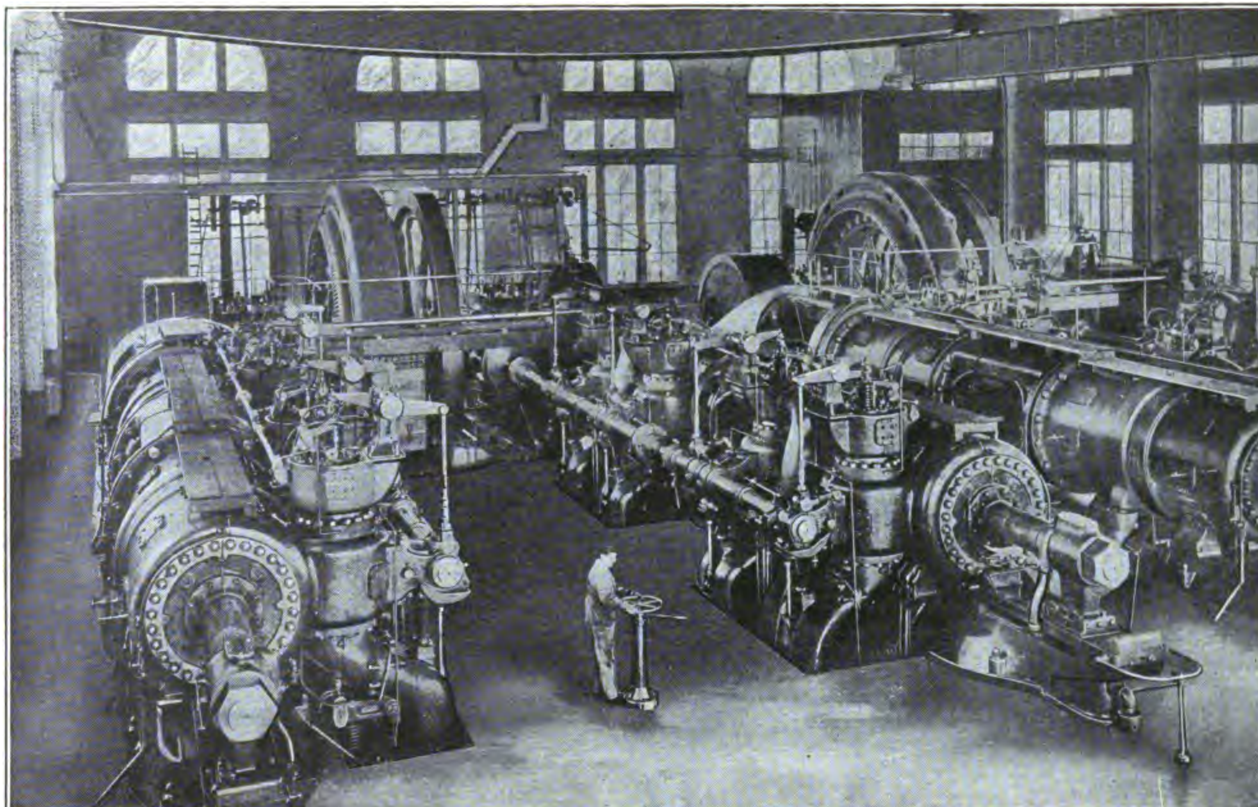
$$\frac{\textit{Ind. ft.-lbs. per cycle}}{778 \times \textit{B.t.u. of combustion}} = \textit{indicated efficiency.}$$

The brake conversion efficiency is

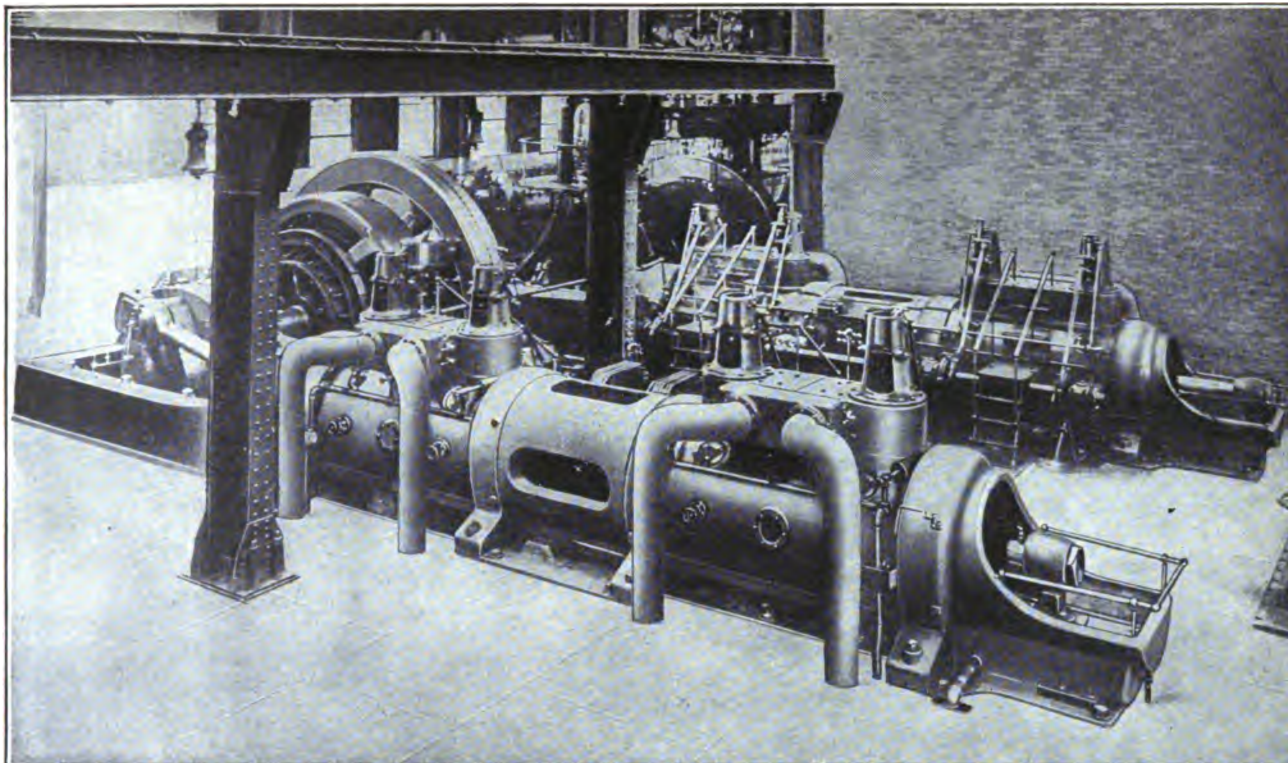
$$\frac{\textit{Ft.-lbs. per cycle at shaft}}{778 \times \textit{B.t.u. of combustion}} = \textit{brake efficiency.}$$

The "B.t.u. of combustion" is equal to the number of heat units per cubic foot or pound of fuel, multiplied by the number of cubic feet or pounds of fuel per minute or hour taken in by the engine and divided by the number of explosions per minute or hour. That is:

$$\frac{h'' \times n}{N_x} = \textit{B.t.u. of combustion,}$$



REAR VIEW OF TWIN TANDEM DOUBLE-ACTING GAS ENGINES IN THE SAN MATEO POWER HOUSE OF THE CALIFORNIA GAS AND ELECTRIC CORPORATION. THE BORE OF THE CYLINDERS IS 42 INCHES AND THE STROKE IS 60 INCHES. THE ENGINES DRIVE ALTERNATING-CURRENT GENERATORS AT 88 REVOLUTIONS PER MINUTE, AND THESE GENERATORS ARE OPERATED IN PARALLEL WITH THE CORPORATION'S HUGE SYSTEM. THE FUEL IS WATER GAS OF ABOUT 600 B.T.U. PER CUBIC FOOT, MADE FROM CRUDE OIL. THE FULL-LOAD COMPRESSION IS 95 POUNDS, ABSOLUTE. EACH TWIN ENGINE IS CAPABLE OF DELIVERING 4600 BRAKE HORSE-POWER CONTINUOUSLY.



TWIN TANDEM DOUBLE-ACTING GAS ENGINE AT THE McKEESPORT PLANT OF THE NATIONAL TUBE COMPANY. THE CYLINDER BORE IS 32 INCHES AND THE STROKE 42 INCHES. THE ENGINE DRIVES A DIRECT-CURRENT GENERATOR, OPERATED IN PARALLEL WITH SEVERAL STEAM-DRIVEN GENERATORS. RUNNING ON BLAST-FURNACE GAS OF 80 TO 90 B.T.U. PER CUBIC FOOT. THIS ENGINE DEVELOPS FROM 1600 TO 1800 BRAKE HORSE-POWER REGULARLY, WITH FREQUENT HEAVY OVERLOADS. THE COMPRESSION PRESSURE IS 180 POUNDS PER SQUARE INCH, ABSOLUTE. REGULATION IS EFFECTED BY VARYING THE QUANTITY OF GAS ADMITTED.

in which

$h''$ = B.t.u. per cubic foot (or pound) of fuel, as delivered to engine,
$n$ = Number of cubic feet (or pounds) of fuel delivered to the engine per minute,
$N_x$ = Number of explosions per minute.

The mechanical efficiency is the ratio of brake to indicated efficiencies or horse powers, as one may prefer. Thus,

$$\frac{\textit{Brake efficiency}}{\textit{Indicated efficiency}} = \textit{mechanical efficiency},$$

or

$$\frac{\textit{Brake horse power}}{\textit{I.H.P.}} = \textit{mechanical efficiency}.$$

There is no way of ascertaining the conversion efficiency of an internal-combustion engine except by actual test. The theoretical cylic efficiency of the four-stroke cycle is given by the formula:

$$\textit{Cyclic efficiency} = 1 - \frac{1}{r_c{}^{n-1}},$$

but this cannot be applied to an actual engine because it is based on adiabatic compression, combustion, and expansion, and complete instantaneous combustion at the moment when the crank is passing the dead center.

The formula for efficiency, assuming regular loss and gain of heat during compression and expansion respectively, is

$$\textit{Indicated efficiency} = \frac{P_x\,K_e - P_c\,K_c}{\frac{h}{1+a}\,(r_c - 1)},$$

but this cannot be considered generally accurate for the same reason that the equations for net work and pressure, which contain $K_c$ and $K_e$ or $K_c$ alone, are not always accurate.

The only method of determining the efficiency of a gas or oil engine with any approach to accuracy, therefore, is to make a series of tests of the machine. Even this procedure does not always give accurate results, because engine indicators are seldom sensitive enough and are usually afflicted with lost motion in the pencil mechanism; because it is difficult to maintain uniform operating conditions throughout a period sufficiently long to measure the fuel consumption, and because it is almost impossible to maintain the quality of the fuel constant when the fuel is a gas. The composition of natural, coke-oven, producer or blast-furnace gas is likely to change several times during a short run. Pressure producer gas is more nearly stable than any of the others just enumerated because the holder tends to equalize the quality of the gas, and can be manipulated so as to supply an absolutely uniform gas throughout a very brief run.

It is possible to obtain a supply of absolutely uniform gas of any kind for a short test by providing a holder large enough to contain all the gas required for the test, filling it before beginning the test and not putting any more gas in it until the test is finished. This is a rather expensive and inconvenient method, however, and is impractical for making field tests.

# INDEX